MODERN

STEAM TURBINES

EDITED BY

ARTHUR R. LIDDELL

VOL. I.

THE SCHULZ STEAM TURBINE

MODERN STEAM TURBINES

BRITISH AND FOREIGN.

COMPRISING DESCRIPTIONS OF SOME
TYPICAL SYSTEMS OF CONSTRUCTION.

UNDER THE EDITORSHIP OF

ARTHUR R. LIDDELL.

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Modern steam turbines

THE

SCHULZ STEAM TURBINE

FOR

LAND AND MARINE PURPOSES

With special reference to its application to War Vessels,

BY

MAX DIETRICH,

Marine-Oberingenieur A.D. of the German Navy,

With 43 illustrations and diagrams, and 6 tables.

NEW YORK
E. P. DUTTON AND COMPANY
31 WEST TWENTY-THIRD STREET
1907

PREFACE.

The growing importance of the steam turbine for land purposes, and still more for the propulsion of ships, has prompted us to bring some of the results of British and foreign endeavours in this field within the reach of interested parties.

The volumes of this series comprise descriptions of some of the systems of which encouraging practical trial has been made, or which, in view of the past record of their authors, have a *prima facie* claim to be taken seriously as engineering probabilities of the future.

It is clear that a new departure in practical engineering must present for solution many new problems, each of which may be approached in a variety of ways, and a knowledge of the successes and failures of other designers who have made practical trial of this or that expedient will doubtless supply hints in many directions. Even untried proposals may point the way to solutions of difficult questions, whether of principle or of detail.

In regard to matters of controversy between rival inventors, which show themselves in some of the works in the collection, it is not the intention to take sides between different engineers or different schools of engineering, nor to show preference in any direction, but simply to lay *ex parte* statements before our readers

with all impartiality, in the hope that mutual understanding and appreciation may thereby be promoted, and that the advantages of different systems may, in course of time, be combined in one good standard type of engine.

It is hoped that the publication of the present series may prove a timely one and may bring advantage to the engineering profession.

ARTHUR R. LIDDELL.

August, 1906.

This volume is an authorised translation of "Die Dampfturbine von Schulz," by Max Dietrich, published by C. J. E. Volckmann, of Rostock, German 1906.

THE SCHULZ STEAM TURBINES.

These turbines are not only for *land purposes*, such as that of supplying the motive power for light and power installations and locomotives, but are also especially adapted for *marine work*, such as the driving of the screw propeller.

Richard Schulz, the designer of these turbines, as Engine-works Manager of the *Germania Shipyard of Friedrich Krupp in Kiel*, has had many years of experience in the construction of engines for use on land, and still more in that of marine engines. He has, therefore, had ample opportunity of studying the conditions which such turbines must fulfil, if they are to enter into successful competition with the ordinary reciprocating engine. Since in the *Germania Shipyard* at Kiel not only large torpedo boats, but the very largest ships of the German Navy are built, Herr Schulz has been obliged to consider the suitability of the *turbine for warships of every possible kind.*

The patents published hitherto and the designs embodied in them show that both *the action* and *the reaction principles* have had application, and have been developed as independently as possible. Turbines on the *action principle* with *expansion stages*, in which each of the working-wheels runs in a separate chamber the walls of which support the guide blades, have been known for more than thirty years (See "*Roues et turbines a vapeur,*" *by K. Sosnowsky*, Paris). The steam turbines of *Rateau*, *Zoelly*, *Curtis*, &c., are newer constructions on the same principle; but no types hitherto known make complete use of the available motive power—in this case the steam—*when the work to be done is small.*

To alter the output, *Zoelly* simply throttled the steam. *Rateau* sometimes, and *Curtis* probably always, employed as constant an *initial pressure* as possible, but devised no means of regulating the pressure in the subsequent expansion stages. Consequently, they failed to make full use of the steam energy, especially when the output was inconsiderable.

In the new *Schulz turbines*, on the other hand, not only the initial pressure, but the pressure in every expansion stage is so regulated, either by hand or by some automatic device, that full use can be made of the steam, even for a minimum of work.

The principal patent in this connection is the "*Regulating apparatus for multi-stage expansion turbines*," No. 132,868; class 14c, of March 26th, 1901. This and the patents mentioned in the specification refer to

(a) Several expansion stages, as in the designs of *Rateau*, *Zoelly*, &c.

(b) Several expansion stages with subsidiary speed-expansions, as in those of *Curtis*, *Riedler*, *Stumpf*, &c.

The *new action turbine* of *Schulz* (Fig. 14) is devised chiefly for use on land. It has from one to ten or even more expansion stages, each of which can be amplified by one or several subsidiary speed stages. In every detail of its construction this turbine shows the requisite simplicity and strength. Between the fixed and the rotating portions are clearance spaces of from $\frac{1}{8}$ to $\frac{1}{4}$ of an inch. This ensures safe working.

A special valve or ring slide regulates the work done in each expansion stage, so that the available boiler pressure is used to the best advantage, whatever may be the output.

This condition is fulfilled likewise by the *Laval* turbine; for in it, whether the work done be little or much, the pressure and change of temperature of the steam are utilized as completely and evenly as possible.

Schulz' action turbines can be used both for the smallest outputs of 2 or 3 horse-power and for the highest up to 15,000 horse-power or more. Moreover, both high and low peripheral velocities down even to 65 feet a second or less can be obtained. The number of expansion stages, and especially also that of the subsidiary speed stages, depends on the required peripheral velocity. The shaft may be placed either vertically or horizontally.

Schulz' reaction turbine differs from the well-known Parson's turbine principally in the fact that the end thrust unavoidable in this sort of turbine is nullified without the employment of a grooved "labyrinth piston." *Schulz* has solved the problem in many different ways; the contrivances for this purpose are described in detail further on.

This reaction turbine is remarkable for its strength and for the ease with which it can be constructed. Its length is less than that of turbines of other systems; and since in spite of this the steam is utilized satisfactorily, this turbine is powerful in proportion to its size and weight (See the comparison between the various turbines in Figs. 39 to 43). It can be employed for outputs of from 50 to 15,000 horse-power, or even more.

Superheating is considered an advantage in both kinds of turbine.

The first problem attacked by *Schulz* was to make turbines reversible, that is, to enable them to drive ships either forwards or backwards. He laid the greatest stress on this point in his first designs, which go back as far as the year 1897. Only in his later plans did he pay increased attention to the other properties which should be possessed by an engine for use on land or sea. The most important of these is that the turbine should be economical, not only

when its output is large, but also when the work done is moderate or little.

For a marine engine, economy in the use of steam is usually demanded only for motion ahead. When the engines are reversed, a rapid answer of the ship to the rotation of the propeller is of more importance.

The engines of warships or merchant vessels are reversed when the ship is manœuvring or is going in or out of a port, and when the danger of a collision or some similar unforeseen circumstance arises. In these cases an economical use of the steam is generally not necessary, for when the engines are worked irregularly, we cannot regulate the generation of steam in the boilers so that the pressure is no greater than the needs of the moment require. In most cases superfluous steam is formed, which must be carried off to the condenser to avoid waste of fresh water.

One simple means of obtaining a reversible turbine system is to place two equal turbines on a common shaft, one for motion ahead and the other for motion astern. The weight and space requirements of this system are, however, so great as to render it unsuitable for a marine engine. The designers of steam turbines have often supposed that they could avoid this difficulty by using a large turbine for motion ahead and a small one for motion astern. Experiments on such systems have, however, shown that they are, at any rate, inapplicable to men of war.

The disadvantages of a go-astern engine of small output are not so noticeable in smaller vessels, such as tugs or torpedo boats, but in larger ships they are very much in evidence. In such cases the hull answers to the propeller much too slowly, and it is a long time before the ship begins to move in the direction opposite to that in which she had previously been going, perhaps at a high speed. This drawback is exaggerated by the fact that a turbine engine is obliged to have a comparatively small screw.

If, however, vessels of any sort manœuvre indifferently, they

are in frequent danger of collision, while men of war lose their value as fighting units.

We draw the following conclusions as to the conditions which should be satisfied by a marine turbine. For motion ahead it must make full use of the available energy, so as to economise coal. For motion astern this is of less importance, but the turbine should be of a size sufficient to affect the ship's motion promptly and effectively.

Schulz has made a number of designs to satisfy these conditions, and has protected them by patents in Germany and elsewhere. In these the chief consideration has been to make the go-astern turbine as powerful as possible without its at the same time being unreasonably extravagant in the use of steam. Although in manœuvring a large quantity of steam must be at our disposal, and often some of it must be led away into the condenser or into the open air, yet occasionally this superfluous steam is not available, as for instance, when the engines are reversed for several minutes at a time. It follows that a go-astern turbine should not use more steam than would have been required during the time in which it acted for motion ahead.

It is convenient to enclose the go-ahead and go-astern turbines in a single case. *Parsons* also recognised this fact, and a year later than Schulz he took out a patent for a go-astern reversing turbine placed abaft of the go-ahead engine. He thereby economised weight and space, but the reversing turbine had too small an output and was not sufficiently accessible for inspection and repairs.

The two first *Schulz* turbines (Figs. 1 and 18) were turned out from the *Engineering Works* of the *Germania Shipyard* at *Tegel*, near Berlin, in the years 1898 and 1900. The improvements which time has brought were partly tested on these experimental turbines.

In 1901 the second of the turbines represented in Figs. 18 and 19 was, immediately after completion, placed in a boat of 19 tons displacement and performed all its manœuvres without a hitch. A description of these turbines is given further on.

The first turbines designed by *Schulz* are shown in Figs. 1 and 2. They were patented in April, 1898, as "*Compound turbines*" (No. 103,879; Class 14). Many experiments on the driving of screws and dynamos were made in 1899 with the radial turbine of Fig. 1. Some results of these tests are given in Table I. The maximum output was 51 electric horse-power, when the number of revolutions was 1,700 per minute and the boiler pressure was 205lbs. per square inch.

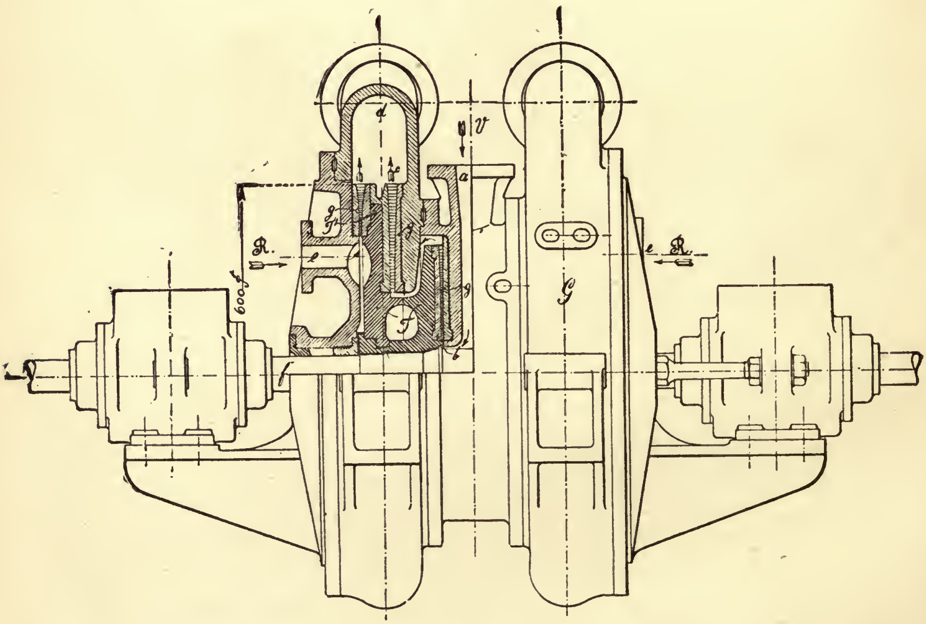


Fig. 1.

In the case *G* are one or more working wheels fastened to the shaft *f*. These carry on their sides the working-wheel blades arranged in concentric rings. Attached to the case are the fixed

blades g, also concentrically arranged, which guide the steam in the necessary direction to the working-wheel blades. Part T of the working-wheel drives forwards: the other part, T', backwards. The entry of the steam is regulated by a three-way valve. The tube V leading from this opens into the steam chest a and enables the steam to drive forwards. The steam finds its way to the ring of blades b, lying next to the shaft f, and after going through the other rings in order, finally passes through the outermost ring c and leaves the case at d. Thence it passes into the condenser or to the open air. The tube R, also leading from the three-way valve, opens into the two steam chests e, and enables the steam to drive backwards. In

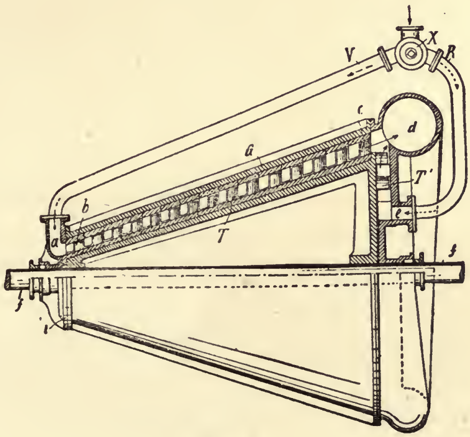


Fig. 2.

this case also the steam passes first through the blades nearest the shaft and finally, after its energy has been exhausted, leaves the case at d, as before.

The end thrust is eliminated by the arrangement of two equal turbines with the common steam supply for motion ahead arranged between them and with separate supplies for motion astern arranged at their outer sides. One of the oldest of *Schulz'* patents, viz., the engine illustrated in Fig. 2, may here appropriately take its place.

It is composed of a go-head turbine with axial flow and a go-astern radial turbine. The working-wheel here takes the form of a drum shaped like a truncated cone. The two ends of this cone are closed by the plates i and i', which serve as bearings for the shaft f. The case g, which supports the fixed blades, fits the shape of the working parts. The rings of working-wheel blades (b to c), which drive forwards, are fixed on the sloping sides of the working drum. Those which drive backwards are placed on the base of the drum. The design and arrangement of the steam supply may be seen in the figure. The rings of blades for motion astern are of as large a diameter and are situated as near the steam exit d as possible, so that they may offer no resistance while the engine is driving ahead.

A disadvantage of reaction turbines is the end thrust with which the steam acts on the working-wheel. In large turbines this may reach very considerable values. *Parsons* eliminates this thrust by the use of counteracting pistons, forming the well-known "labyrinth apparatus."

These increase the length of the turbine and consequently waste space; moreover they are extravagant in steam. Economy of space is, however, one of the chief aims of modern builders of every kind of engine, and loss of steam (though this is less important) should also be avoided.

The use of reaction turbines as marine engines necessitates the employment of a large number of fixed and revolving blades, if the peripheral velocity is to be sufficiently low. Marine turbines must, therefore, have comparatively many more rings of blades than stationary engines. It is necessary, especially for marine purposes,

that these numerous blade rings should be divided among several drums, so that the shaft may, by reason of the shortness of the individual drums, be more firmly and securely mounted. *Schulz'* designs satisfy this condition adequately and at the same time eliminate the end thrust.

Schulz patented a *compound steam turbine* in November, 1900 (No. 137,792; Class 14^c), in which two turbines are symmetrically arranged with their end thrusts acting in opposite directions. A number of rotating blades are fastened on two drums of different diameters. By means of a suitable flow of steam the end thrusts of these drums are opposed in such a manner that the pressure on the thrust block is entirely eliminated, or at least brought within reasonable limits.

Fig. 3 shows a compound turbine of this kind, in which the diameters of the rings of fixed and revolving blades (as in Fig. 2)

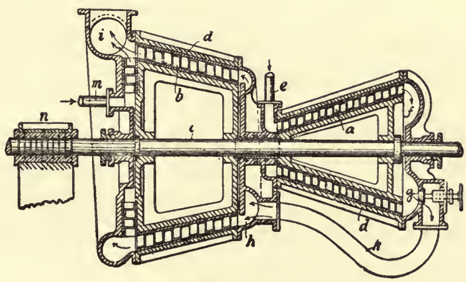


Fig. 3.

increase in the direction of flow of the steam. In the patent, however, provision is also made for the case in which the rings of the high and low pressure turbines respectively are of equal diameter.

In Fig. 3 the high-pressure turbine with the smaller blade rings is denoted by the letter *a*, the low pressure turbine with larger rings by *b*. Both revolving drums are enclosed in a common case *d*

and are fixed to the shaft *c*. The steam enters the turbine at *e*, and after passing through the fixed and revolving blades of the high pressure engine, is regulated by the valve *g*. It then goes through the connecting pipe *f* and the tube *k*, finds its way at *h* into the low pressure turbine, and is led away at *i* into the condenser or into the open air. In *a* and *b* the steam flows in opposite directions.

The fixed blades in *a* and *b* are placed in such a way that both *a* and *b* drive in the same direction. The valve *g* at the mouth of the pipe *f* regulates the pressure of the steam as it leaves *a*. In this way the resultant end thrust on the thrust block *n* can be altered within certain limits.

As in Fig. 2, the moveable reversing blades are fastened to the base of the low-pressure working drum and the fixed blades to the corresponding cover of the case. The steam for reversing enters through the pipe *m*, and in this case also reaches the smallest of the concentric rings of fixed blades first. The steam, as in the case of direct rotation, leaves at *i*, close to the outermost ring.

This radial go-astern turbine can also be replaced by a shorter axial engine, as is shown in a diagram in the patent specification.

Compound turbines are composed of two or more turbine-drums, whose rotating parts are placed in separate cases, and sit either on a single shaft or on several distinct shafts. Of the many combinations which are given in the before-mentioned patent No. 137,792, we shall mention only two.

Fig. 4 shows four turbine-drums *a*, *b*, *u*, *v*, placed on a single shaft. The arrows show the direction of the driving steam and

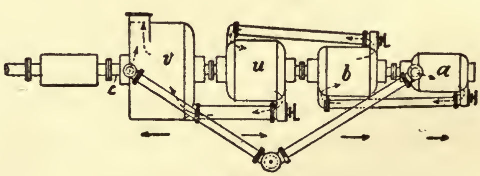


Fig. 4.

hence the directions of the corresponding end thrusts. We see that *a*, *b*, and *u* give an end thrust to the shaft in a direction opposite to that given by *v*.

In Fig. 5 the drums are divided between two shafts. Here we see an arrangement in which the smaller turbines *a* and *b* act on one shaft, the larger ones *u* and *v* on the other. The distribution of the steam flow necessary to counteract end thrust is clear from the figure.

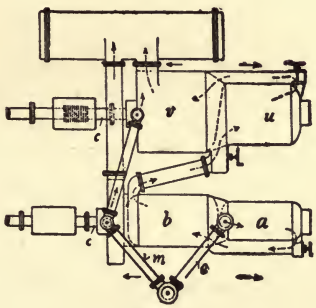


Fig. 5.

In this diagram only the grouping of the four drums is new. *Monsieur Tournaire* (as is well known) had as early as 1853 described in detail a method in which several turbines, worked one after another by a flow of steam, can be made to drive several shafts simultaneously.

In stationary turbines the end thrust should, of course, be abolished as completely as possible. In marine engines, on the other hand, the end thrust should be adjusted by a suitable choice of the diameter of the revolving parts, so as just to counterbalance the thrust of the screw. When the ship is going ahead, this screw thrust is towards the bow, so that the turbine thrust should be of equal amount, but should act in the direction of the stern.

For better regulation of the axial thrust the use of manometers on suitable parts of the high and low pressure engine cases is advantageous. When the pressures of the steam in the two turbines are known, the end thrust may be determined. The difference of pressure and hence the thrust may be altered to some small extent by the valve *g* (Fig. 3).

Another design patented in July 1901, (No. 135,937; Class 14*c*) also aims at the abolition of end thrust. Two such turbines are represented in Figs. 6 and 7.

Fig. 6 shows a longitudinal section of a turbine rotating in a single direction only. It is admirably suited for use on land. The flow is partly radial and partly axial. The axial portion of the rotating parts is denoted by *a*, and the radial by *b*. The case *e* is

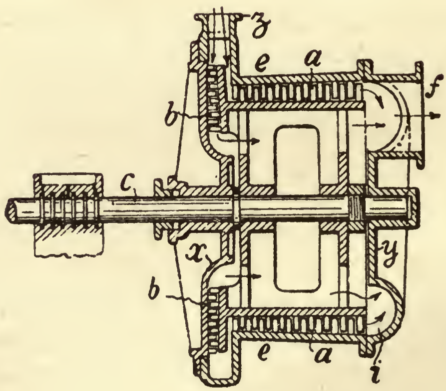


Fig. 6.

closed by the covers x and y. The radial portion of the revolving blades is placed in a ring-shaped extension of the case, and here also is the steam chest z for the entering steam. The cover y carries the chest f, through which the steam exhausts. The direction of flow of the steam is shown by arrows.

Fig. 7 shows a section through a turbine with radial and axial flow, which is reversible. For direct motion this turbine

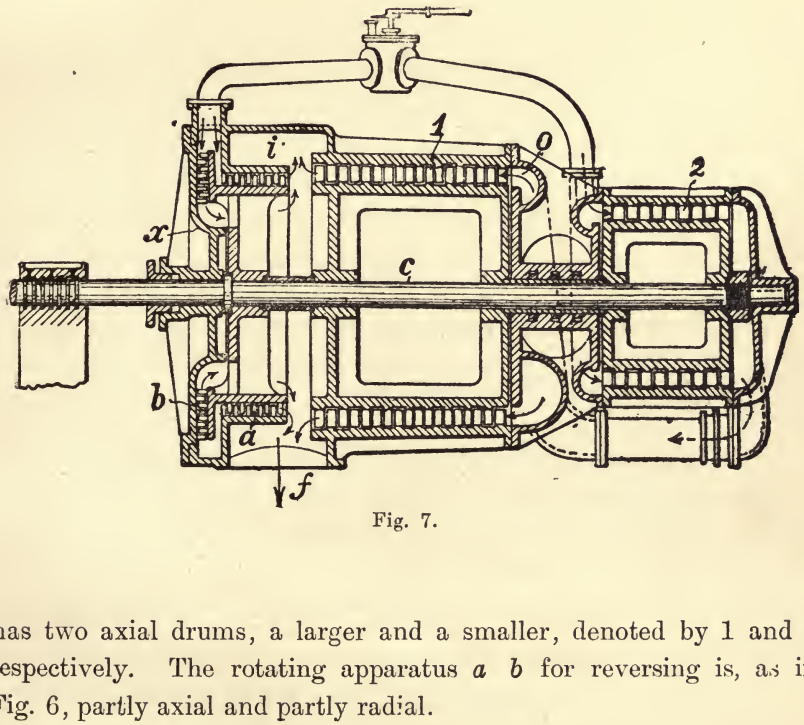


Fig. 7.

has two axial drums, a larger and a smaller, denoted by 1 and 2 respectively. The rotating apparatus *a b* for reversing is, as in Fig. 6, partly axial and partly radial.

In the turbine of Fig. 6, and in the reversing portion of Fig. 7, the steam is divided on its entrance into the case and passes at the same time through the radial (b) and the axial (a) parts of the common

rotation rings. Both *a* and *b* give an end thrust towards the right, but the steam flowing against the back of the flange-like part *b* before its entrance into *a*, gives a thrust towards the left. This latter may be regulated at pleasure—so, for instance, as exactly to counterbalance the thrust to the right.

In eliminating the end thrust, the chief consideration is, in the go-ahead turbine, the difference of pressure in 1 and 2—in the reversing turbine the breadth of the flange placed on the drum wall of *b*.

After solving the problem of the design of a turbine which should fulfil the necessary conditions of ease of reversal and elimination of end thrust, *Schulz* turned his attention to the question of economy.

Steam turbines with several blade rings work economically only while they are giving the largest possible output under maximum boiler pressure.

The cross section of the steam passage is generally calculated to suit the greatest output, and the steam is led into the first blade ring at as high pressure as possible. When the steam goes through the other rings its pressure gradually diminishes till the condenser is reached, when, provided the energy be well utilised, it becomes comparatively small.

If a smaller output be demanded from the turbine with its steam passages formed as above, it becomes necessary to resort to throttling. By this, however, the steam loses its pressure to a considerable extent even before it reaches the first ring, and economy is thereby sacrificed. To meet this objection, *Schulz* devised means

of varying the cross section of the steam pipes in each separate ring of blades. The entry of the steam to the revolving parts is thereby so regulated with reference to the desired output, that the full boiler pressure is employed, whatever may be the rate of working. This is effected by the placing of a ring-slide valve before each ring of guide blades or before a portion of the ring (German Patent No. 132,868; Class 14c; March, 1901).

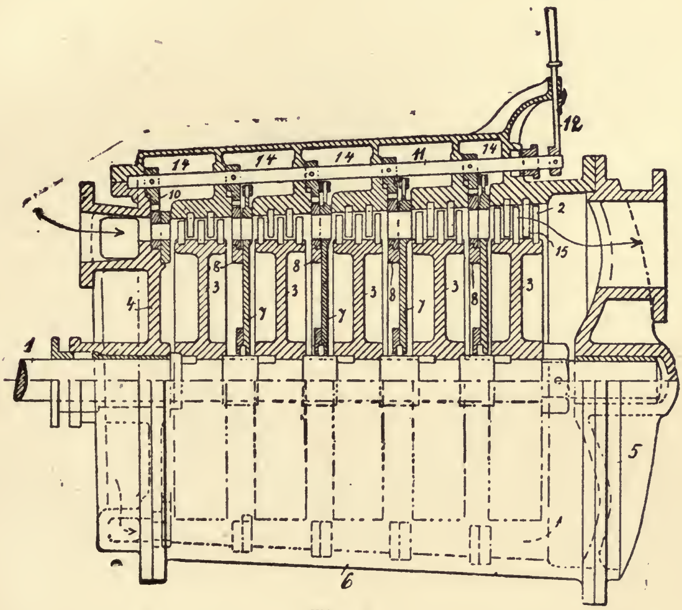


Fig. 8.

By altering the position of this valve we can leave free a varying number of holes in the fixed rings for the passage of the steam to the rotating parts. In Fig. 8 the individual fixed rings are stationary, but the ring slides, while arranged for simultaneous adjustment by a common lever, can on occasion be moved separately. The same regulation of the steam may also be obtained by the moving of the individual blade rings relatively to their ring-slide valves.

There is no essential alteration in the operation of the steam when the rings of guide blades are made moveable, either separately or simultaneously, instead of the ring slides.

Fig. 8 shows a section of a turbine constructed in this manner. In Figs. 9 to 12 the above-mentioned adjustments are depicted.

The action turbine of Fig. 8 has axial flow and is provided with five expansion stages. Each of these has three subsidiary

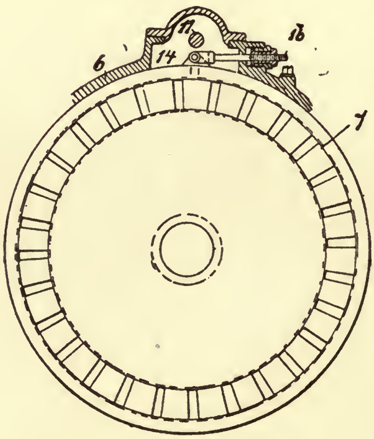


Fig. 9.

speeds. The case (6) has a greater diameter at the steam exit than at the entrance, but the working-wheels all have the same mean diameter.

The five wheels (3) are fastened to the common shaft (1), which is supported by the covers (4 and 5) at the ends of the turbine. Each wheel bears three revolving blade rings (2), corresponding with the speed stages. The first fixed blade ring of each stage is adjustable, and is placed on the corresponding partition wall. The two others are formed of blades (15) arranged in ring fashion round the

inner periphery of the case. On the cover (4) is the chest for the introduction of the steam; on the cover (5) is the chest for its exit. The direction of the steam flow is shown by arrows. The different expansion stages are formed by the partition walls (7) (Fig. 9). They reach nearly to the shaft, and the clearance space being so small, no appreciable loss of steam can there take place. The ring slide (8) and the corresponding adjustments are represented in Fig. 10.

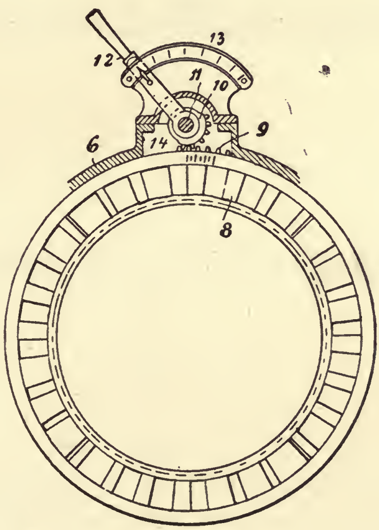


Fig. 10.

Fig. 11 is an expansion drawing, showing the relative positions of the holes in one of the ring slides and in the corresponding partition wall when the output is at a maximum and when it is at a minimum respectively.

As shown in Fig. 10, the ring slide carries on the upper part of its circumference a number of teeth (9), fitting into the toothed wheel (10), which can rotate about the axis (11). The slide valve is moved by the adjusting-lever (12), and its position is given by the

scale (13). The divisions through which the axis (11) is moved are denoted by 14. The adjusting apparatus on the partition is shown in Fig 9. The adjusting-rod (16) and the spindle (11) pass through ordinary stuffing boxes.

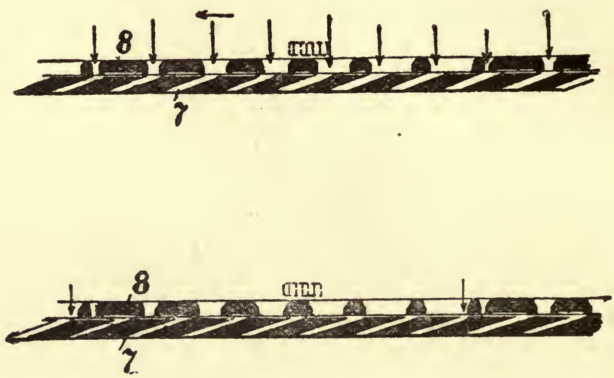


Fig. 11.

In this turbine the partitions have 28 entrance holes or nozzle-like channels placed at equal intervals and having the same cross-section. The ring slides have also 28 openings which, though alike in each quadrant, vary in size and distance apart. In Fig. 11 two such quadrants are shown developed on to a plane.

The distances between the openings are so chosen that in the extreme position all the seven steam holes are open and the corresponding expansion stage in question undergoes complete impingement. If the valve be moved in the direction shown by the arrows, the holes on the left of each quadrant are closed. If the valve be moved further still, into the position shown in the lower half of Fig.

11, the steam only enters through one hole in each quadrant. Hence a turn of the lever 12 through one division always opens or closes four holes.

By means of the partition walls and ring slides of Figs. 9 and 10 we may, therefore, obtain either a complete or a partial flow of steam. We can, however, partially cut the steam off by another method, as is evident from Fig. 12. This latter device has the advantage that the steam from leading channels or nozzles, which are arranged close to one another, is directed against the working-wheel in question in a single stream.

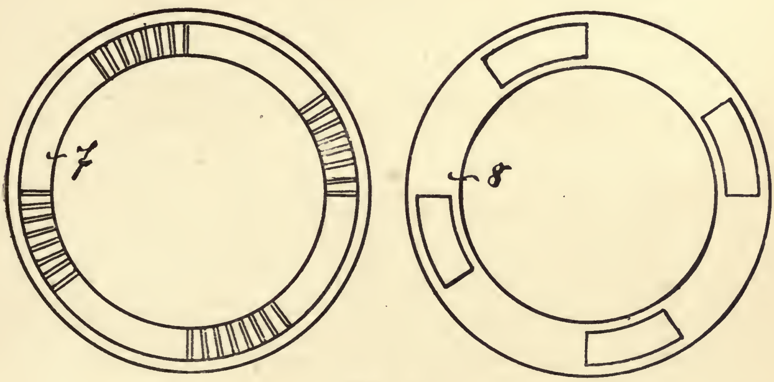
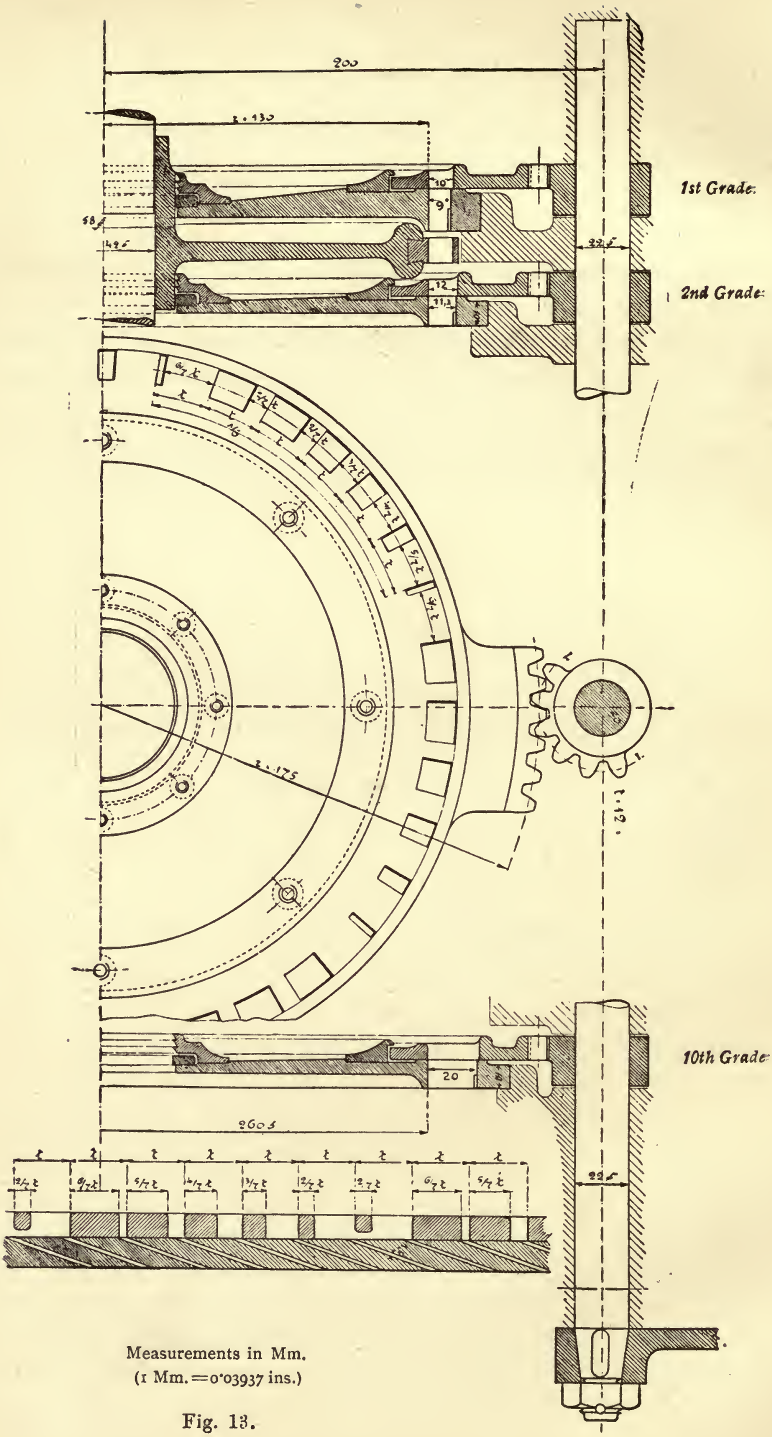


Fig. 12.

If the ring slide is to be worked automatically, the lever (12) is connected with a powerful governor. The partitions are adjustable separately, as before stated. We are, therefore, able to regulate the width of the steam passage in each partition and hence for each expansion stage separately. If, for example, all the holes are open, we can by moving one or other of the partitions, increase the pressure of the steam leaving the corresponding expansion stage, and so secure the best economy for the desired output.



Measurements in Mm.
(1 Mm. = 0·03937 ins.)

Fig. 13.

Fig. 13 shows the construction of the ring slides and partitions, and their positions in the turbine. The openings in them are also shown in detail.

To prevent the steam from passing along the shaft from one expansion stage to the next, the corresponding borings in the walls are edged with saw-like teeth. The steam vortices and the friction so caused reduce the loss of steam considerably. In addition, a special packing-ring is provided at each partition.

The advantages of *Schulz' new designs* have been already recognised by many German firms, who are now building turbines on his system.

A larger type for outputs of from 500 to 800 horse-power and 2,000 revolutions per minute, is given in Fig. 14 This turbine has four expansion stages, each with two subsidiary speed stages. In regard to the division into pressure and speed stages, every other possible variation of these may here be made. Also, the steam flow may be either axial or radial, and may be partial in some of the stages and complete in others. An example of a radial turbine of three expansion stages and three subsidiary speed stages, with special regulating apparatus and radial flow, is shown in a diagram of the above-mentioned patent No. 132,868. The preference, however, in this case given to axial flow.

In the turbine of Fig. 14, the driving steam is sent through the fixed and moving blades very effectively. The widths of the holes and the number of these in the various partitions, as well as the number of expansion and speed stages, are in all cases adjusted

to the requirements in regard to available boiler pressure, number of revolutions per minute, &c.

Fig. 8 shows a similar turbine with five expansion stages, each having three speed stages.

The action turbine in Fig. 14 is designed for a stationary plant. It will, however, serve just as well for warships and merchant vessels. It will give both large and small outputs with high or low peripheral velocities, and will work at the highest pressures. It should be especially noticed that the efficiency of these turbines is not sensibly diminished even when the output and peripheral velocity are small. It is only necessary that, with due consideration to the conditions in each case, the expansion stages be fairly numerous. Under all circumstances, however, *Schulz'* action turbines will not need more than the minimum number of blades and wheels usually adopted for the purpose in hand, so that the cost of construction is not excessive.

In the turbine shown in section (Fig. 14) the regulation of the steam in the first expansion stage is effected by a number of small stop valves. In the other expansion stages the steam supply is controlled by ring slides, which are placed immediately in front of the groups of nozzles in the partitions (see Fig. 12). Thus the cross-section of the steam passages is suited to the work required. The ring slides may be adjusted by a cogged wheel system, by worm gearing, or by levers. The governor is placed on the free shaft wheel near the stop-valve and the apparatus for moving the ring slides; these latter are adjusted by a very simple device. The governor can be arranged so as to adjust both the ring slides and the stop valves, or it may be made to act only on the latter.

The turbine can be divided horizontally into equal halves. It is made of cast iron or cast steel, or, in very small engines, of bronze. The guide wheels or partitions are made of the same

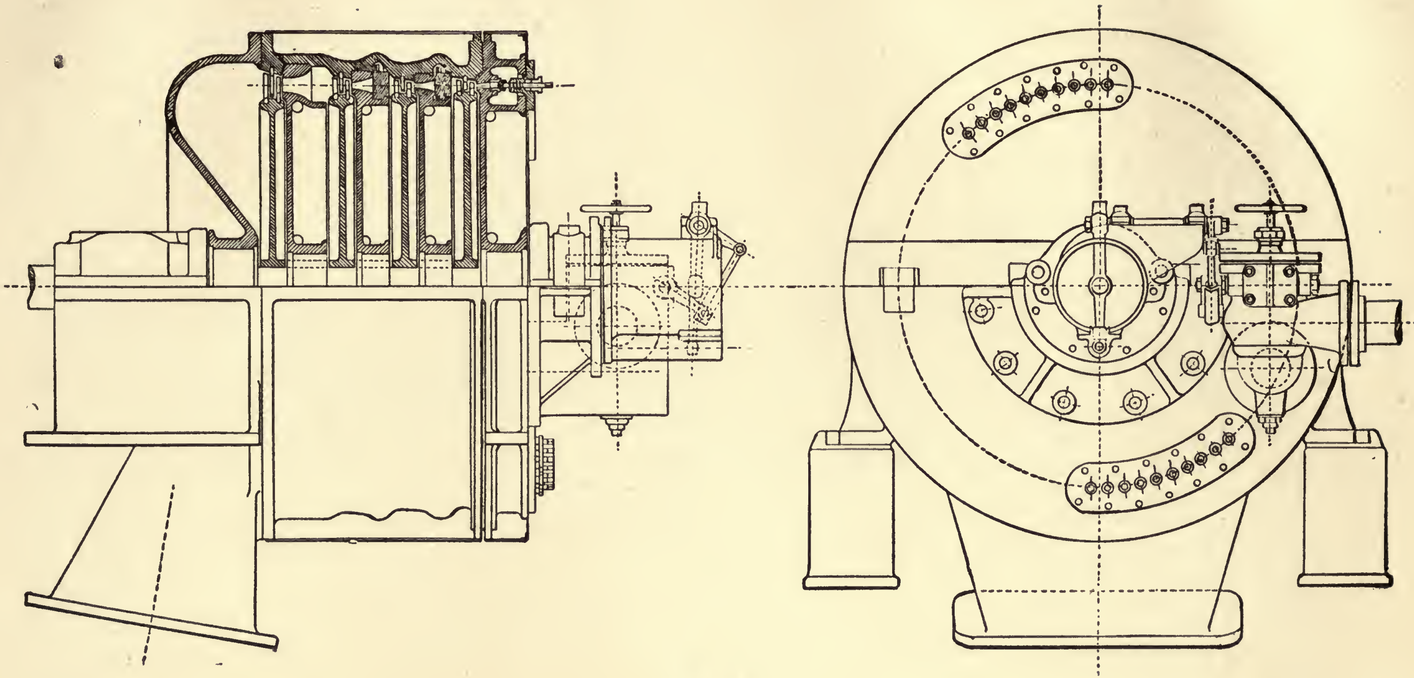


Fig. 14.

material, and are placed steam-tight in the case. The working-wheels are of steel, accurately turned on the lathe and well balanced. The revolving blades are of delta-metal, and are strongly and simply attached to the wheel ring by a specially contrived method. Comparatively wide clearance spaces are left between the fixed and the rotating parts, and even when highly superheated steam is employed, no trouble due to the rubbing of the blades against each other or to such like causes can arise.

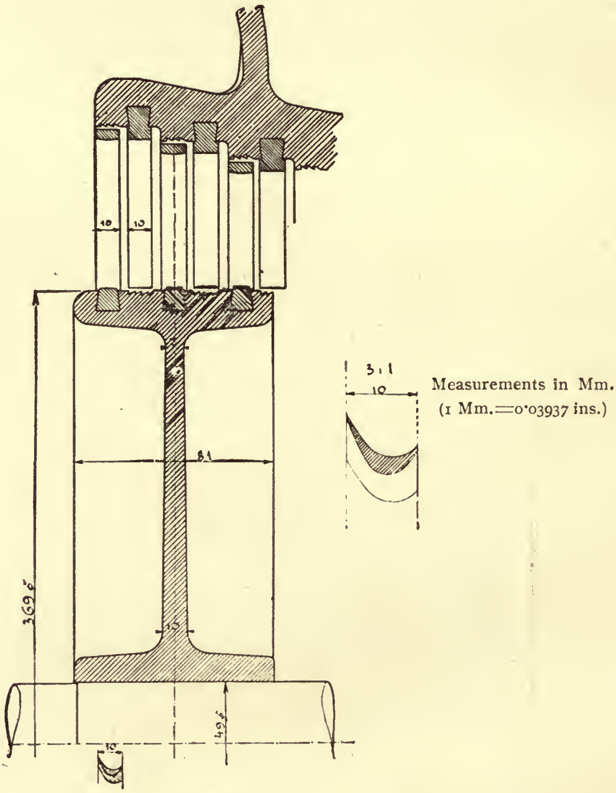


Fig. 15.

Fig. 15 shows the method of attaching the fixed and moving blades in the older turbines of the year 1900. These were dove-tailed into suitable grooves and fixed firmly by a special method.

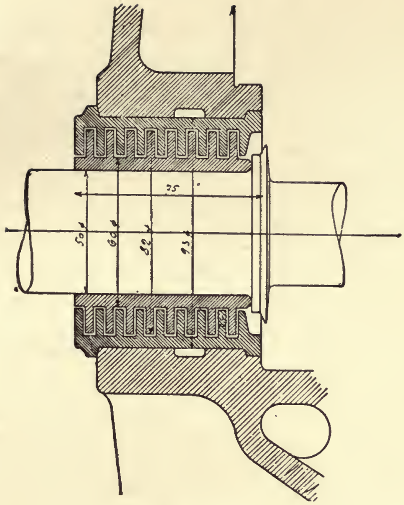


Fig. 16.

Measurements in Mm.
(1 Mm. = 0·03937 ins).

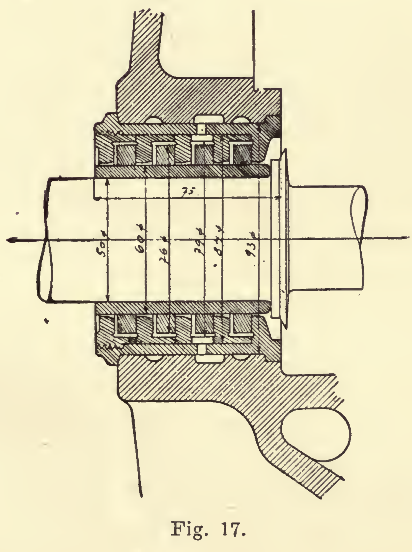


Fig. 17.

Such fixing is not effected in the more modern types by the caulking tight of the distance pieces, since it is difficult in this case to avoid a twisting of the blades. Besides, if the positions of the blades be altered they cannot be recovered.

Here also we see the saw-like grooves, which create vortices in the axially flowing steam. These tend to prevent loss at the clearance space between the case and the guide blades and between the latter and the working-wheel. In recent times, however, less stress has been laid on this point. The labyrinth apparatus closes the shaft at its exit from the case.

In Figs. 16 and 17 are shown the constructions of the older labyrinth stuffing boxes, which closed the shaft of the *Schulz* turbine. These stuffing boxes are extended for a considerable distance along the shaft, so as to make the path of the steam in the labyrinth as long as may be. Fig. 16 shows one of these stuffing boxes, in which the labyrinth is formed by two cylindrical castings fitting into each other with rings screwed to them. The inner casting is fastened to the shaft by means of an arrangement of screws, placed axially. The labyrinths may be thus regulated, so that the spaces left on one side of the rings are as little as $\frac{1}{750}$ inch, while those left on the other side are comparatively larger and can serve as steam-chambers.

Fig. 17 also shows a packing of *Schulz*' design. Here the stuffing box is composed of a number of well-fitting rings. All these are placed on a common cylindrical casting and are kept in place by a ring at the end. The labyrinths are here formed by the peculiar cross section of the individual rings. *Schulz*' new labyrinths differ from these types chiefly in the shape of the rings screwed to the casting.

As in all action turbines, the steam pressure in each expansion stage is the same on both sides of each working-wheel. A considerable end thrust, such as we find in multi-stage reaction turbines is

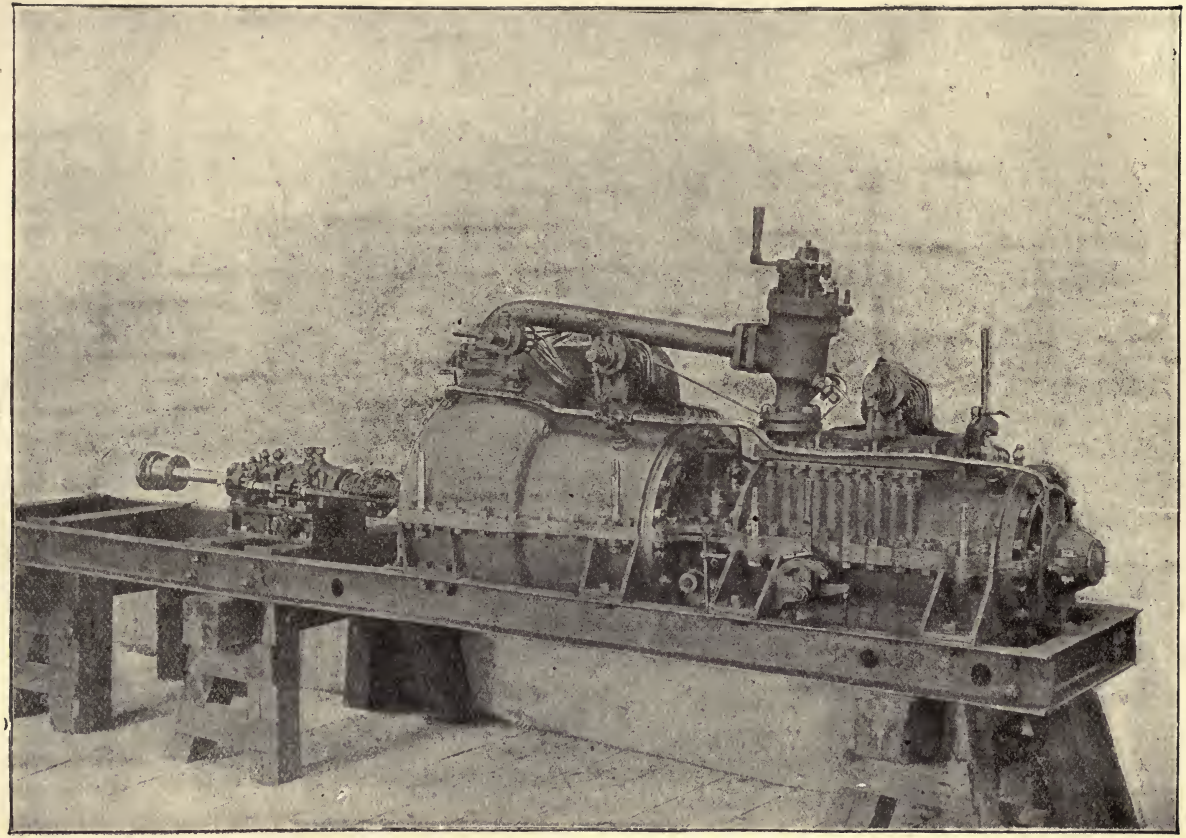


Fig. 18.

in this case impossible. Counteracting pistons and similar devices can, therefore, be dispensed with.

We see, then, that the new *Schulz* turbines satisfy all requirements; they are reliable in working, simple in design, and very economical in use of steam, employing the full available pressure for every variation in the output.

Many crucial experiments have been made on the two steam turbines mentioned on page 11 (Figs. 1, 18, and 19), which were built in the works of the *Germania* in *Tegel*. The second, especially, has been repeatedly tested, both in the workshops and in a boat of 19 tons displacement on *Lake Tegel*, near *Berlin*. This experimental boat was built principally to test the manœuvring powers of the new turbines. In December, 1901, soon after the engine was placed in the boat, it was inspected by His Excellency Admiral *von Tirpitz*, Secretary of State for the German Admiralty, and went through all its manœuvres without a hitch.

This turbine is shown in Figs. 18 and 19. Fig. 18, which shows the external appearance of the turbine, is reproduced from a photograph taken in the experimental room of the workshop. On the high pressure case may be seen the adjusting appliances for the partitions (Fig. 9). Within certain limits, these regulate the steam pressure in the various expansion stages. Mention may also here be made of an indicator appliance, which enables graphic representation to be made of the pressure in the various expansion stages. In Figs. 21 to 24, several diagrams of this kind are shown.

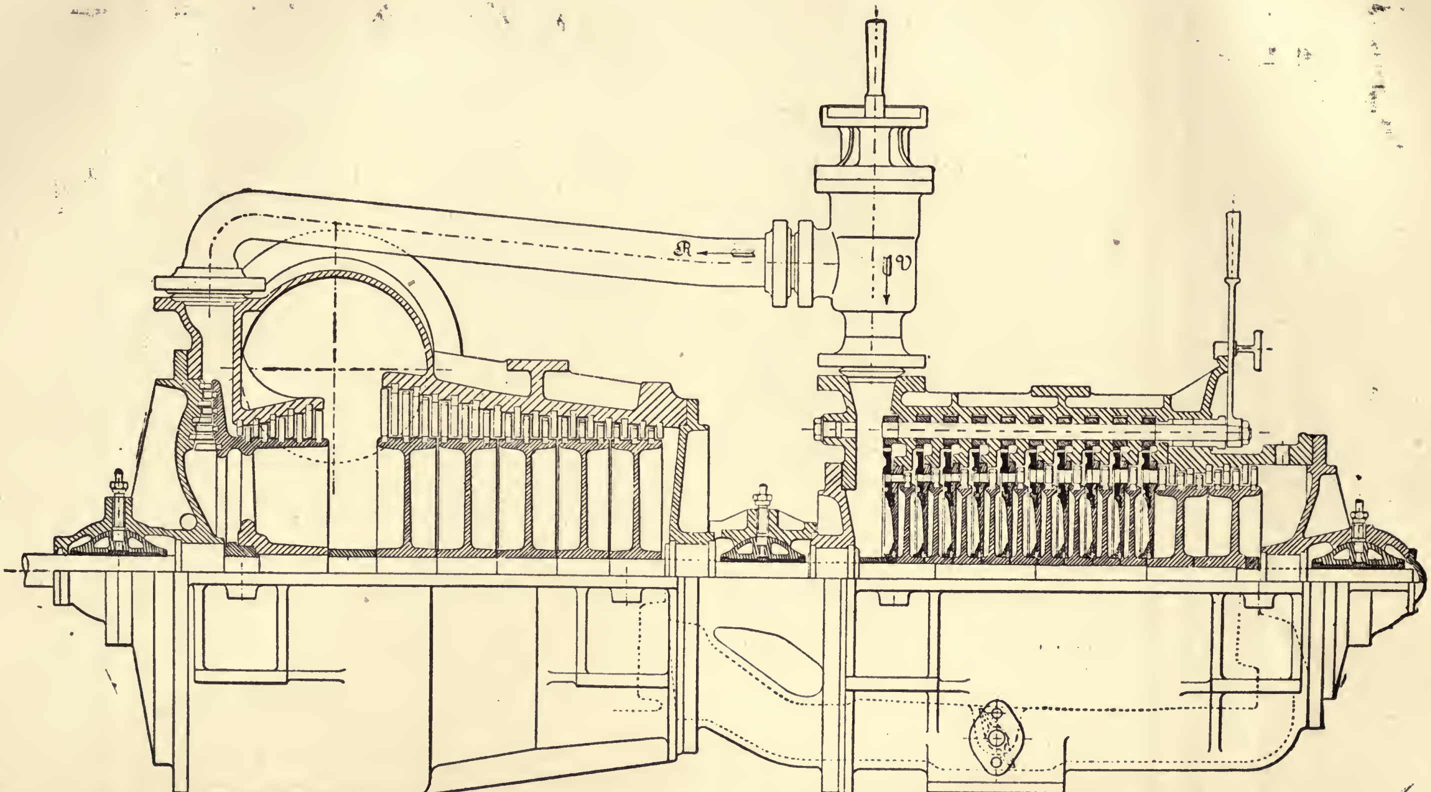


Fig. 19.

The apparatus required for this purpose consists of an indicator stop-cock to which a large number of tubes (in this case 15) are attached (see Fig. 20). In Fig. 18 three of these cocks are visible. With each expansion stage of the turbine one of them communicates by means of a narrow pipe, so that, when its cone is turned suffi-

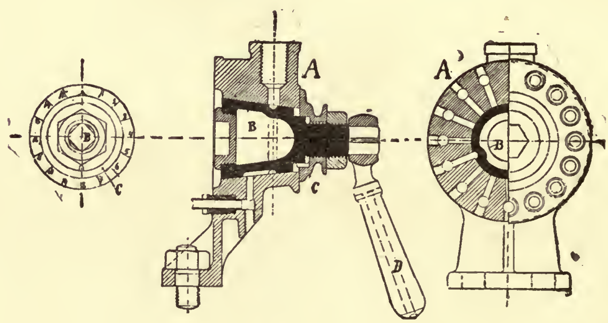


Fig. 20.

ciently and a simultaneous rotation of the indicating paper cylinder is made, the curves shown in Figs. 21 and 22 are described. The three diagrams of Fig. 22 show that, even when the cross-section of the steam passages is reduced to a sixth of its original size, the full available pressure is employed in the first expansion stage. The abrupt fall of pressure noticeable especially in Fig. 22, II. and III., is due to the temporary throttling of the steam passages in the expansion stages in question.

Indicator Diagram, December, 1900.

Revolutions per minute=2050.

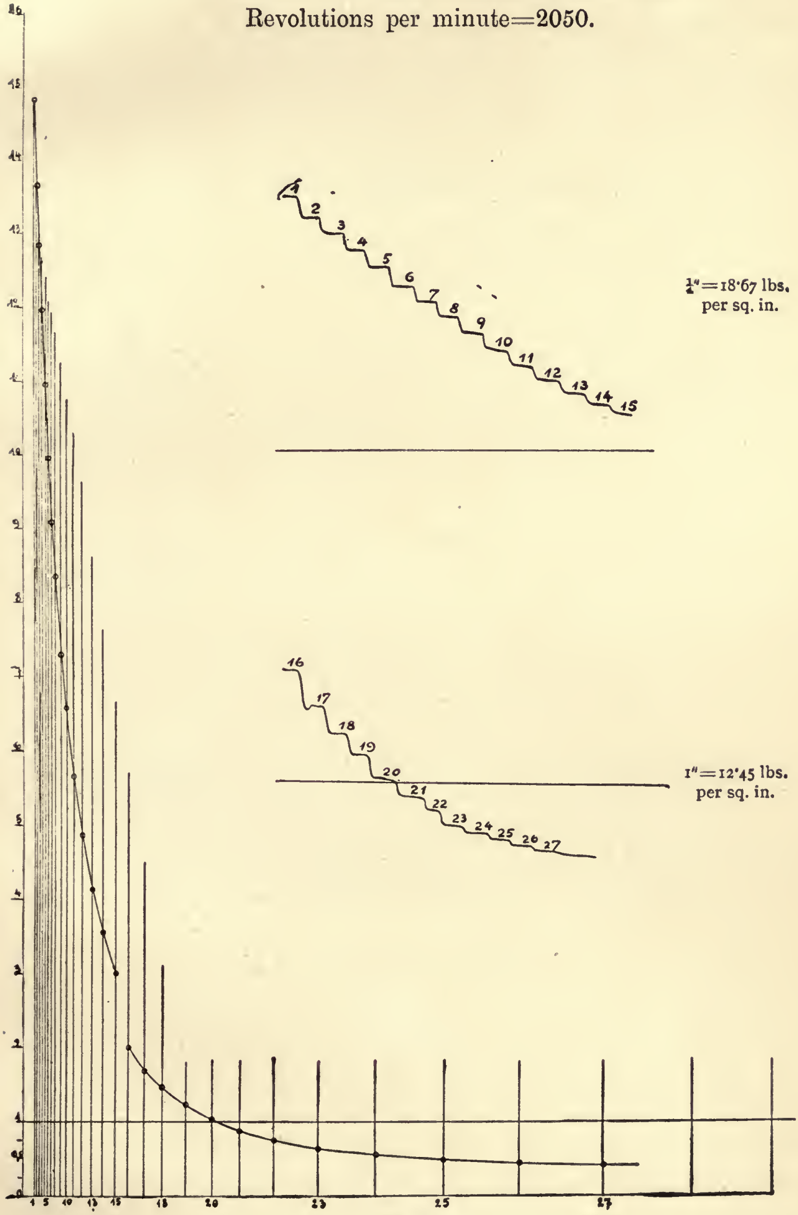
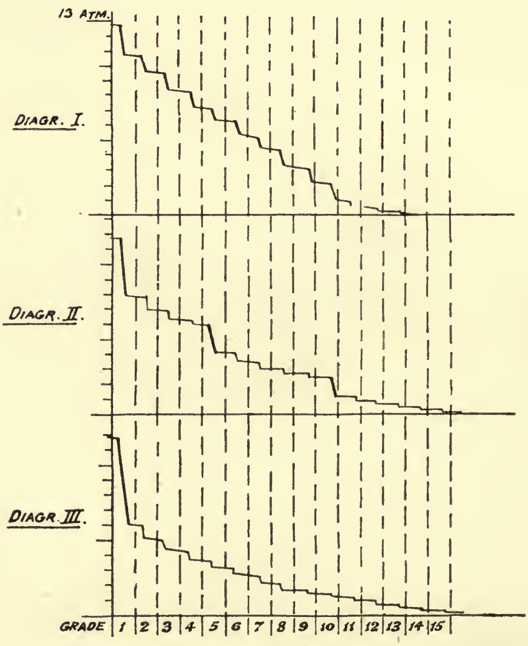


Fig. 21.

Indicator Diagram, January 22nd, 1901.



Grades 1-10
are given $\frac{1}{8}$ passage.
1310 revolutions.

Grades 1, 5, and 10
are given $\frac{1}{8}$ passage.
1440 revolutions.

Grade 1
is given $\frac{1}{8}$ passage.
1440 revolutions.

1 Atmosphere=14·706 lbs. per sq. in.

Fig. 22.

Figs. 23 and 24 show the advantages of placing an indicator on a turbine, just as do Figs. 21 and 22. They were drawn from the pressure curves of the high and low pressure turbines when mounted in the boat and being driven at a diminishing speed. In one case only six passages were open to the steam; in the other 30. The diagrams show plainly how the pressure diminishes in the different expansion stages as the speed decreases, and it is noticeable that, as the output becomes smaller, one expansion stage after another at the exit end of the low-pressure turbine ceases to contribute anything to the work done.

In Fig. 23 six steam passages are open. Of the 30 expansion stages 25 are still in use when the number of revolutions is 890 per minute; 22 are employed when the number is 770; while only 19 are working when the number is as low as 300.

Fig. 19 gives in section the marine turbine of Fig. 18. The high and low pressure turbines are those given in the patent No. 132,868; the go-astern engine is that published in the patent specification No. 135,937 (see Figs. 6 and 8). The details of the design are shown in Figs. 8 to 17.

The direct turbines of this engine are axial; the go-astern turbine is both axial and radial. The high pressure engine is, for the most part, an action turbine; in the low-pressure and go-astern engines the reaction principle is employed. The end thrust is eliminated by the methods already described.

The turbine of Fig. 18 is here illustrated in a tenth of its natural size. It weighs, with mountings and fittings, only a little over a ton, and since its output is 230 horse-power, its weight per horsepower is only about 11 lbs. The case is made of cast iron and is divided horizontally. The working-wheels are of steel and the blades of the well-known delta-metal. The boat used to test this engine was 59 feet long by 9ft. 3ins. broad, and had a mean draught of 4ft. 8ins. The diameters of the various propellers (which were

Indicator Diagrams, January 15th, 1902.

6 Passages in the Pressure Turbine (1-10)..

1 Atmo.=14·7 lbs. per sq. in.

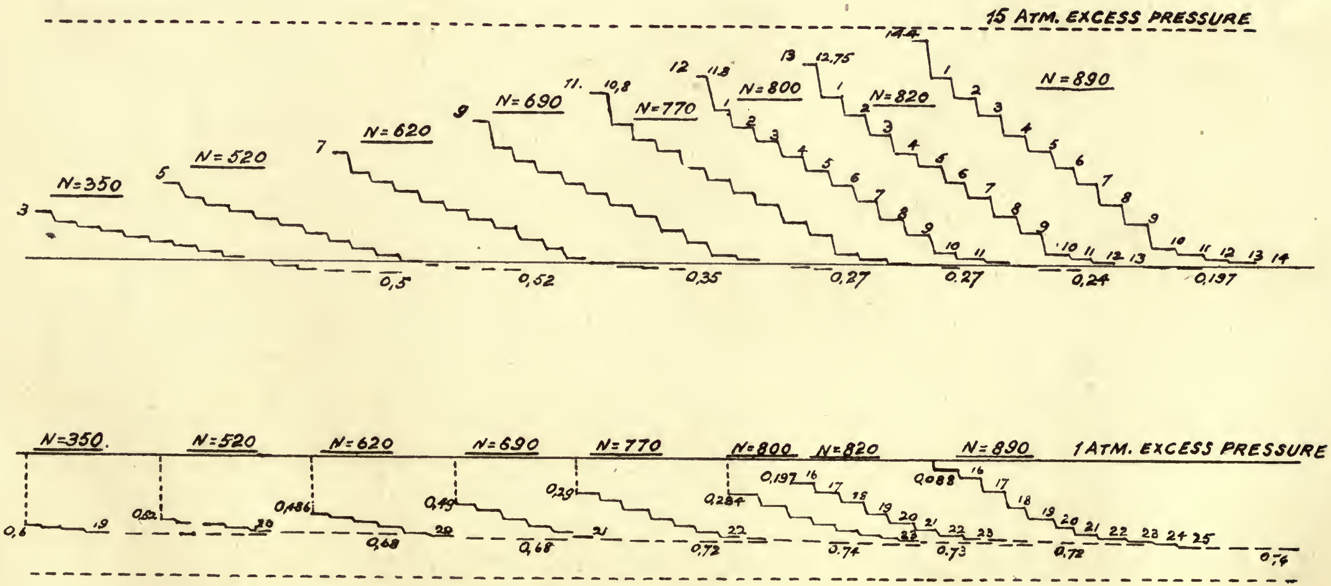


Fig. 23.

Indicator Diagrams, January 15th, 1902.

30 Passages in the Pressure Turbine.

1 Atmo.=14·7 lbs. per sq. in.

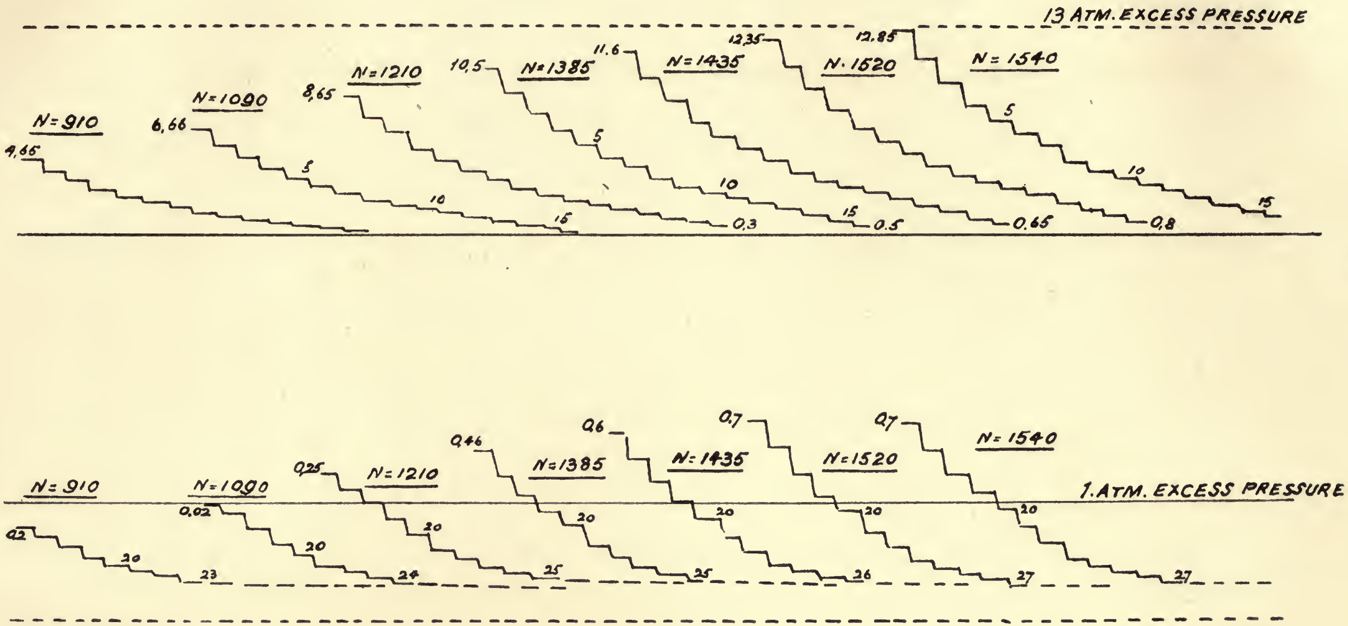
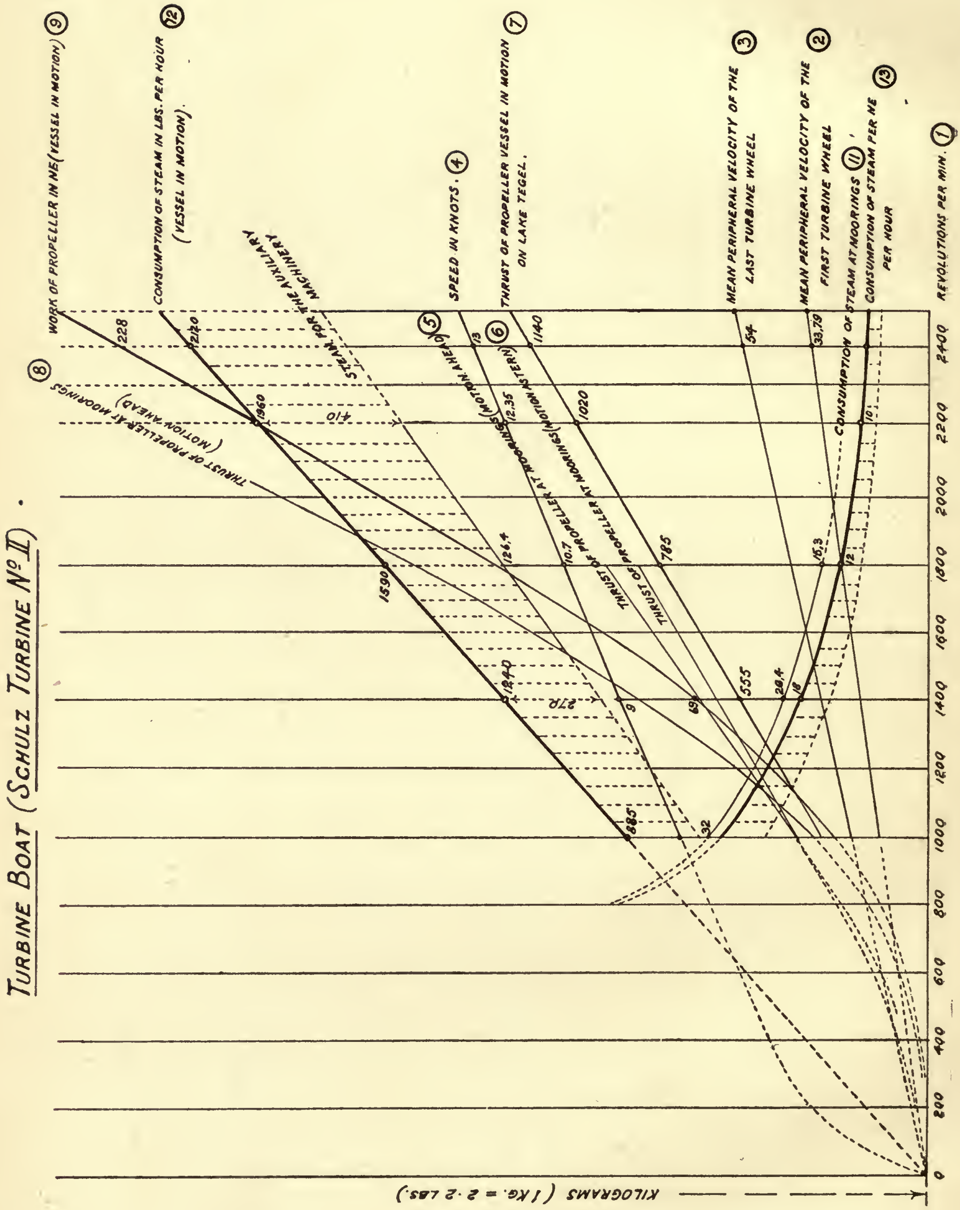


Fig. 24.



TURBINE BOAT (SCHULZ TURBINE No. II).
WORK OF PROPELLER IN NE (VESSEL IN MOTION) 9
CONSUMPTION OF STEAM IN LBS. PER HOUR (VESSEL IN MOTION). 12
STEAM FOR THE AUXILIARY MACHINERY
SPEED IN KNOTS. 4
THRUST OF PROPELLER VESSEL IN MOTION ON LAKE TEGEL. 7
MEAN PERIPHERAL VELOCITY OF THE LAST TURBINE WHEEL 3
MEAN PERIPHERAL VELOCITY OF THE FIRST TURBINE WHEEL 2
CONSUMPTION OF STEAM AT MOORINGS 11
CONSUMPTION OF STEAM PER NE PER HOUR 13
REVOLUTIONS PER MIN. 1
8 THRUST OF PROPELLER AT MOORINGS (MOTION AHEAD)
5 THRUST OF PROPELLER AT MOORINGS (MOTION AHEAD)
6 THRUST OF PROPELLER AT MOORINGS (MOTION ASTERN)
KILOGRAMS (1 KG. = 2.2 LBS.)
0 200 400 600 800 1000 1200 1400 1600 1800 2000 2200 2400

Fig. 25.

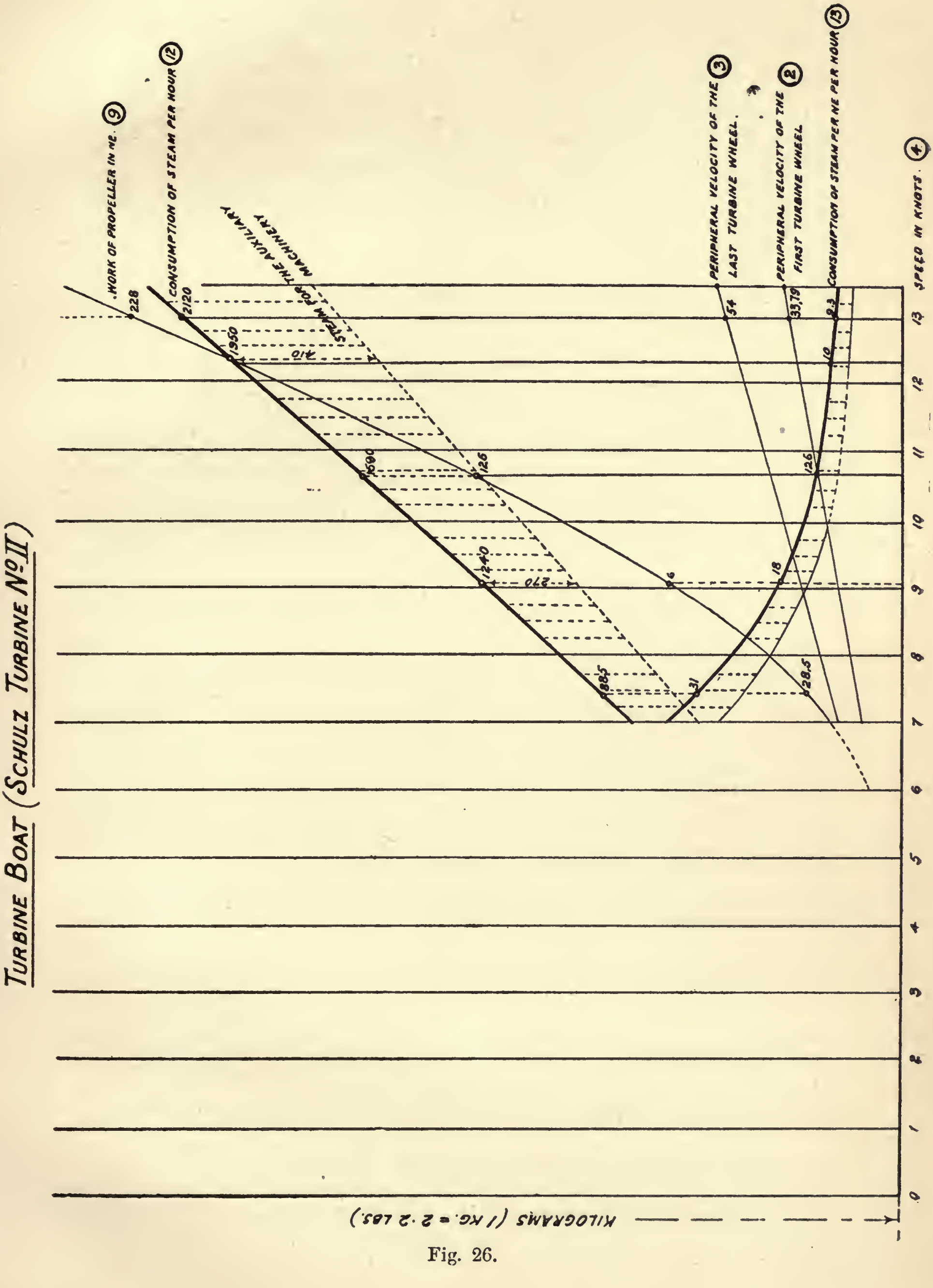

TURBINE BOAT (SCHULZ TURBINE No. II)
WORK OF PROPELLER IN HP. 9
CONSUMPTION OF STEAM PER HOUR 12
STEAM FOR THE AUXILIARY MACHINERY
PERIPHERAL VELOCITY OF THE LAST TURBINE WHEEL. 3
PERIPHERAL VELOCITY OF THE FIRST TURBINE WHEEL 2
CONSUMPTION OF STEAM PER HP PER HOUR 13
SPEED IN KNOTS. 4
KILOGRAMS (1 KG. = 2.2 LBS.)

Fig. 26.

used both singly and in pairs during the test) varied from 12 to 16½ inches, while the number of revolutions per minute varied from 1,000 to 2,200. The greatest speed was more than 13 knots, the highest average over the measured mile on Lake Tegel being 12·35 knots. No vibration was felt even at the highest obtainable speed. The boat manœuvred satisfactorily, and the engines worked perfectly throughout the trials.

The results obtained in the test are given in Table III.

The economy of this comparatively small turbine may be gathered from the curves of Figs. 25 and 26. Especially at speeds above 11 knots a comparatively low consumption of fuel is noticeable. The abscissae in Fig. 25 represent numbers of revolutions; these range from 1,000 to 2,500 per minute. In Fig. 26 the abscissae give the number of knots per hour.

Curve 1 gives the number of revolutions per minute.

Curves 2 and 3 give the mean peripheral velocities of the first and last working-wheels of the turbine.

Curve 4 gives the speed of the boat in knots.

Curve 5 gives the screw-thrust when the boat is moored and the engine is driving ahead.

Curve 6 gives the same when the engine is driving astern.

Curve 7 gives the corresponding thrust when the boat is moving.

Curves 8 and 9 give the work done by the screw when the boat is stationary and when it is in motion respectively.

Curves 10 and 12 give the steam consumption per hour.

Curves 11 and 13 give the steam consumption per horse-power per hour.

The excess pressure of the driving steam at this trial varied approximately from 200 to 213 lbs. per square inch. The superheating of the steam was generally about 160 to 180 deg. F.; in some cases, however, it amounted to more than 780 deg. F. The vacuum was never more than 9·4 lbs. per square inch. A better vacuum could not be obtained, owing to the insufficient output of the air-pump.

When the comparatively small peripheral velocities of the first and last turbine wheels (100 and 165 feet a second) are considered, the results shown by Table III. and by the curves in Figs. 25 and 26 must be considered excellent. With an output of 195 horse-power and a speed of 2,200 revolutions per minute, only 22 lbs. of steam per horse-power per hour were used, in spite of the bad vacuum of 9·4 lbs. per square inch. The brake trials in the Tegel workshop during June, 1901, in which a higher vacuum was at disposal, gave an appreciably larger output. In fact, for every extra lb. per square inch in the vacuum, the turbine gave respectively 13·0, 13·7, and 15·1 extra horse-power when the numbers of revolutions per minute were 2,000, 2,200, and 2,600. Allowing for this, we get an additional output of 240 horse-power when the number of revolutions is 2,200 per minute, while the steam consumed per hour for each horse-power of output is 17·8 lbs., instead of 22 lbs. (see Table IV.)

The steam used during these experiments in the various auxiliary engines was led into the low-pressure turbine, so that its energy could be completely utilised. The steam consumption in these auxiliary engines is not included; it was found by special investigations to be about 880 lbs. per hour. The economy of the turbine is considerably increased if this steam is also reckoned, as in the curves of Figs. 25 and 26, where the steam consumption for the subsidiary engines is denoted by the shaded portion.

According to the latest trials of the "*Turbinia*," fitted with large *Parsons turbines*, the steam consumption with the highest output of 1,600 to 1,700 horse-power is about 20 lbs. per horse-power per

hour. It must be noticed, however, that these turbines are appreciably larger than the *Schulz turbines* under discussion. Moreover, according to page 193 of "*Engineering*," August 1st, 1903, the velocities of the steam in the first and last blade rings of the *Parsons turbine* were 150 and 280 feet per second respectively, the number of revolutions being 2,200 and 3,000 a minute. The velocity in the *Schulz turbine*, on the other hand, was only 100 to 165 feet per second, when the output and speed of revolution were at their maxima.

The results of Table I. apply to the turbine of Fig. 1 : those of Tables II., III, and IV. to the turbine of Figs. 18 and 19.

TABLE I.

Experimental Turbine No. 1 Driving a Dynamo. 1899.

Number of Experiment.		1	2	3	4	5
		Excess pressures in atmospheres.				
Boiler pressure		15	15	11	7·4	5·2
Steam pressure in entrance chamber		12	12	10·5	7	5
Pressure behind first expansion stage		9	8·9	7·8	4·8	3·5
Pressure in passage from 1st to 2nd turbine ..		2	2	1·9	1·15	0·7
Pressure in middle expansion stage of 2nd turbine..		0·25	0·25	0·2	0	0
Number of revolutions per minute		1700	1400	1200	980	800
Dynamo	Voltage	150	160	100	80	75
	Current in Ampères	250	170	200	110	70
Output	Kilowatts	37·5	27·2	20	8·8	5·25
	Electric Horsepower	51	37	27·2	11·97	7·14

The steam turbine weighs, with all the mountings and fittings, about 12½ cwts.

TABLE II.

EXPERIMENTAL TURBINE No. 2. MAY AND JUNE, 1902.

Pressure in front of engine	No. of Experiment. / No. of Revolutions per Minute.	1 / 1600	2 / 2000	3 / 2600	4 / 3000	5 / 4000	6 / 4600	7 / 5000
7	Pressure behind main turbine	1·08	1·08	1·105	1·08	1 09		
	,, in condenser ..	0·71	0·737	0·684	0·074	0·697		
	Horse-power	44	53·5	53·5	50	62·5		
9	Pressure behind main turbine	1·15	1·1	1·12	1·12	1·1	1·09	
	,, in condenser ..	0·73	0·763	0·737	0·731	0·724	0 75	
	Horse-power	63	78·5	84	84	86	104	
11	Pressure behind main turbine	1 38	1·31	1·35	1·35	1·32	1·3	1·3
	,, in condenser ..	0·71	0·763	0·716	0·72	0·737	0 75	0·697
	Horse-power	78·5	100	110	112	129	126·5	121·9
13	Pressure behind main turbine	1·61	1·55	1·6	1·6	1·54		1 5
	,, in condenser ..	0·7	0·727	0·724	0·71	0·737		0·71
	Horse-power	101	118·5	132	141·5	172		165·8
15	Pressure behind main turbine		1·84	1·81	1·8	1·8		1·7
	,, in condenser ..		0·704	0·71	0·724	0·724		0·697
	Horse-power		141·5	161·5	177	204		185·3
16	Pressure behind main turbine	1·88		1·92	1·9	1·9	1 9	1 9
	,, in condenser ..	0·605		0·71	0·717	0·71	0·71	0·71
	Horse-power	115		184	200·5	230	225	226

This turbine weighs, with mountings, fittings, and reversing valve, less than 21·8 cwts.; i.e., less than 11 lbs. per electric horse-power, when this latter is 230.

TABLE III.

Experiment in the Boat with Turbine No. 2.

1	2	3	4	5	6	7	8	9	10	11	12	13
Number of revolutions per minute.	Mean peripheral velocity in feet per second		Boat's speed in knots.	Screw thrust in cwts.			Output of screw in horse-power		Hourly consumption of steam in lbs.			
	of first	of last		when stationary		when moving	when stationary.	when moving.	when stationary		when moving	
	turbine wheel.			forwards.	backwards.				Total.	per horse-power.	Total.	per horse-power.
1000	46·2	73·8	7·4	7·6	7·6	6·8	33·7	28·5	2380	70	1950	68
1400	64·7	103·3	9·1	13·0	12·1	10·9	82·1	69	3700	45	2730	40
1800	83·1	132·8	10·7	18·5	16·9	15·4	150 4	126·5	5070	34	3500	27
2200	101·6	162·4	12·35			20·1		195			4300	22
2300			12·6			21·2		211			4500	21
2400	110·9	177·2	13			22·4		228			4670	20·5

Vacuum = $\frac{2}{3}$ of an atmosphere.

TABLE IV.

Vacuum in lbs. per square in.	1	9	12	13
	No. of Revolutions per minute.	Effective Horse-power.	Steam per hour.	Consumption in lbs per horse-power.
9·4	2,200	195	4,300	22
12·8		240		17·8

After the experiments with the above-mentioned boat had been carried through successfully, Schulz turned his attention to the construction of an engine suitable for all men-of-war, including battleships, cruisers, and torpedo boats.

Merchant ships, whether they be cargo, passenger, or mail boats, steam steadily at their highest possible speed, except, for instance, when the weather is rough or foggy. The construction of turbines for such vessels, therefore, offers no special difficulties. With men-of-war the case is very different; their engines must always be prepared to develop their utmost power, but this output is required but seldom, and then only for short spaces of time. Moreover, in warships it is demanded that the consumption of fuel in proportion to horse-power be the most economical when the output is small or moderate, say from $\frac{1}{15}$ to $\frac{2}{3}$ of its maximum value.

The multi-stage marine turbines constructed by *Parsons* and *Rateau* use appreciably more coal in proportion to the output when the latter is small, than when it is large. On the well-known sea-going torpedo-boats, "Viper" and "Cobra," the consumption of fuel for small outputs was nearly twice as great as in the sister-ships, "Albatross" &c., which were fitted with reciprocating engines.

Rateau and *Parsons* before him have combined reciprocating engines with large-sized turbines for use when only a small output is required. *Parsons*, however, has gone back to pure turbine engines. The steam consumption on the German sea-going torpedo-boat, "S 125," fitted with *Parsons' turbines*, cannot even yet be accurately determined, the air-pump having at the first series of trials, proved unsatisfactory. Several months were then wasted while the latter was being replaced, and since other mishaps have meanwhile intervened to delay the trial trips, it may be still some time before the coal consumption is determined. However, it is fairly evident from the reports on the first trials, that the above-mentioned conditions will not be satisfactorily fulfilled by this engine.

Schulz has now patented another marine turbine (No. 160,863; Class 65a; April 23rd, 1901), which is shown in Figs. 27 and 28. This plant is arranged for a single shaft only. In ships with several screws it is proposed to use one of these engines for each shaft.

In front of the main turbine a number of turbine wheels are placed, which can also be separated into groups. The steam is cut off from these when the output is large, so as to prevent them from

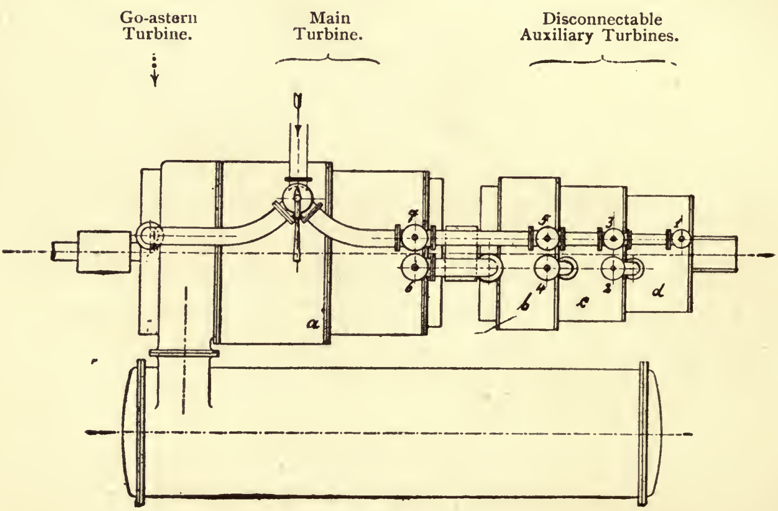


Fig. 27.

sharing in the work. *The smaller the output, the larger is the number* of auxiliary turbines which contribute their share to the work done. When the output is at its smallest, the steam passes through the stop-valve 1 into the turbine *d*, the remaining stop-valves 3, 5, 7, being then closed, and the exit valves 2, 4, 6 opened. When the output is larger, the smallest turbine *d* is cut off by the closing of the stop-valves 1 and 2, and the steam passes by the valve 3 into the

second turbine *c*. On further increase of the work the turbine *c* is also cut off by the closing of the valves 3 and 4; and when the output is at its maximum, all the valves 1 to 7 are closed, so that the steam only enters the main turbine *a*.

By this arrangement an increasing number of turbines contribute their share of work as the output diminishes, so that, owing to the increasing number of expansions, the difference of steam pressure in two successive expansion stages is always very small. The initial pressure may, therefore, be so high that the difference between the pressures at the entrance and exit passages respectively of the engine always maintains its maximum value.

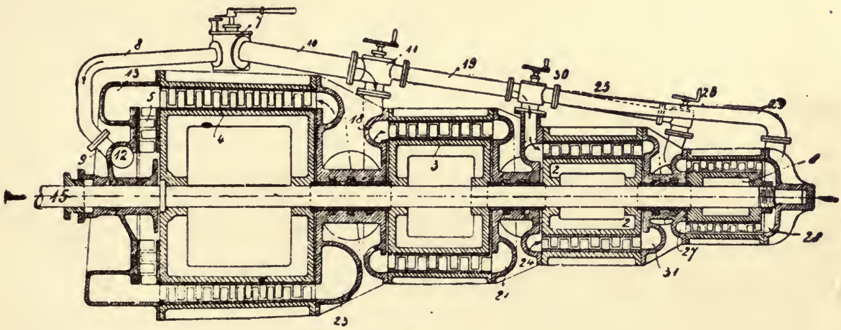


Fig. 28.

If, then, the auxiliary turbines be of suitable size, the full energy of the steam is utilized, whatever may be the speed, and due economy is observed.

It is indifferent whether the whole apparatus is enclosed in a single case or whether each separate turbine has its own cover.

Fig. 28 shows a section of this compound engine. The rotating wheels 1 to 4 are placed on the common shaft 15, and work directly; the wheel 5 is for motion astern. To regulate the end

thrust the steam flows in opposite directions through the largest turbines (4) and the smaller ones (1, 2, 3). The main stop-valve 7 allows the steam for forwards motion to flow through the pipe 10 and the valve 11, through the pipes 10 and 19 and the valve 30, or through the pipes 10, 19, and 25, and the valve 26, into the various turbines. The steam for reversing passes through the pipes 8 and 9. The pipe 29 allows steam to pass from turbine 1 to turbine 2. The circular passages, 18, 23, 24, 27, and 12, serve as chests for the steam before its entrance into the various turbines; the passages, 28, 31, 21, and 13, serve as exit chests. The steam is led to the condenser through the pipe 13.

The *Parsons turbine* is at present the most popular for marine purposes. It is, therefore, of interest to compare this engine with the *Schulz turbine*, especially as lawsuits have been brought by the former engineer against the latter for infringement of patent. These lawsuits were decided in favour of the defendant.

The plaintiff relied chiefly on the English patent, 11,223/97, with which the German patent No. 103,559 corresponds. He maintained that *Schulz*' arrangement had been already protected by this patent and that consequently *Schulz*' patent No. 160,863 (Figs. 27 and 28) was invalid.

Parsons' arrangement in the patent No. 103,559 is shown in Figs. 29, 30, and 31. Though he has made many laborious and costly experiments to get a low steam consumption in both low and high outputs, he has failed, and *Schulz* was the first to obtain a satisfactory solution of the problem. Parsons subsequently followed in the path already trodden by Schulz.

In *Parsons'* attack on *Schulz'* system (Figs. 27 and 28) attention was first called to an article by the naval engineer, *Herr Grauert*, in the *Marine-Rundschau*, of January, 1904. Notice was also taken of Grauert's remarks in "Steam Turbines," by *Dr. A. Stodola*, relating to the economy of steam necessary for warships.

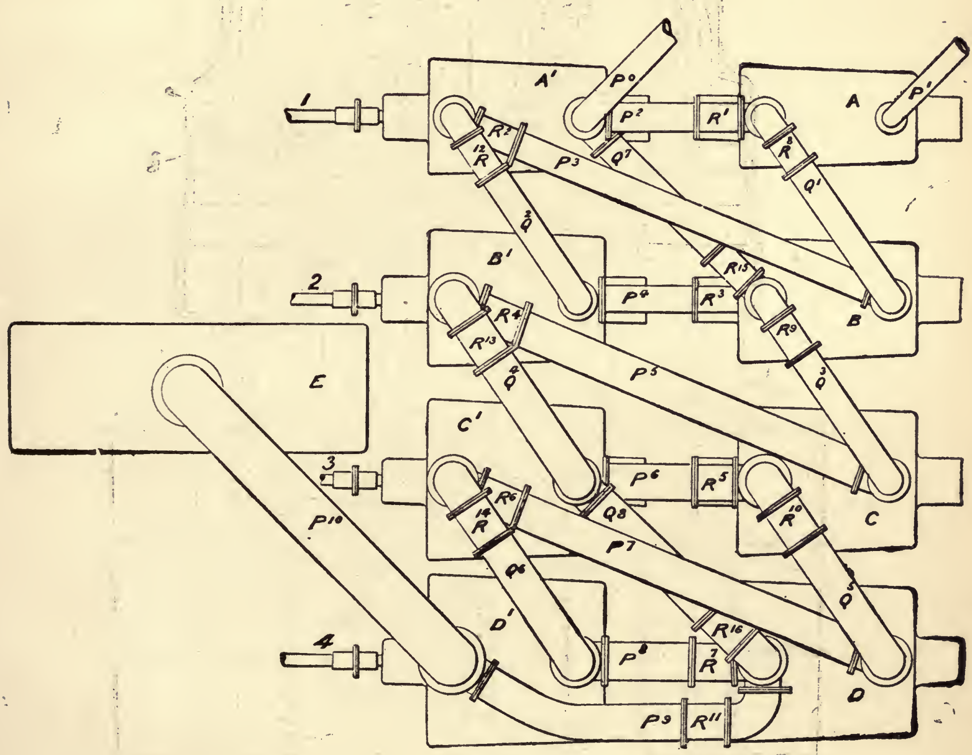


Fig. 29.

Now in this article we read :—

"For constant output, the fuel burnt per horse-power is appreciably higher at low rates of revolution. If, however, the output and speed of rotation fall simultaneously, the consumption of steam is altered but little."

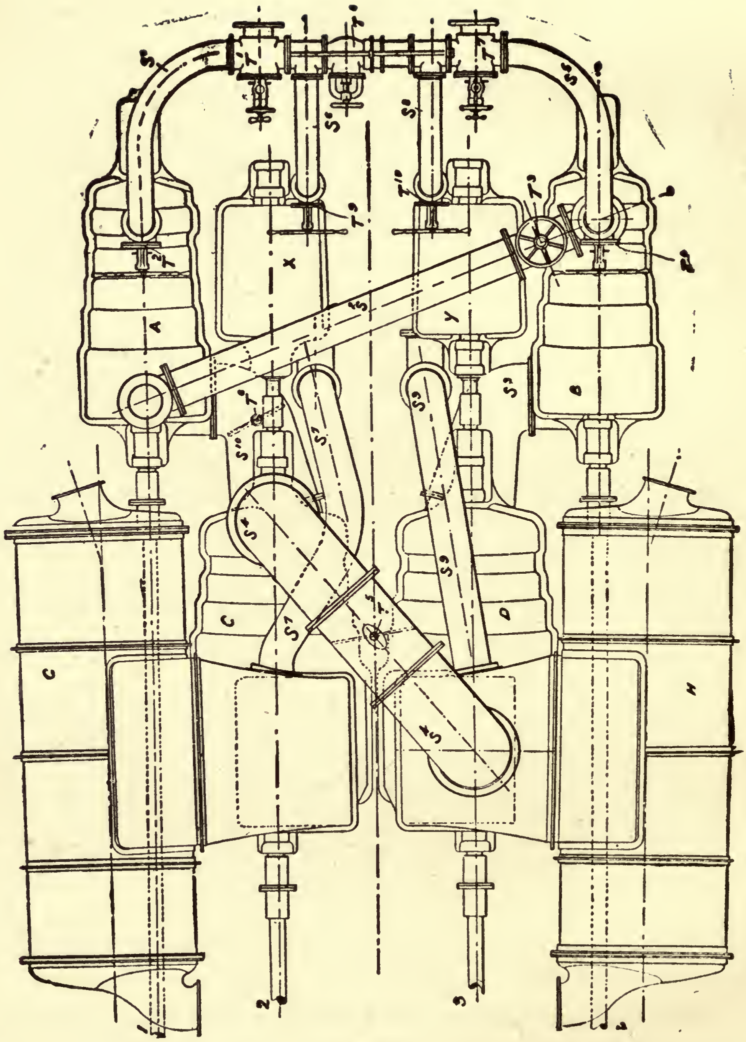


Fig. 30.

The first of these two statements would be intelligible only if it were, under normal circumstances, possible for the output of a marine engine to remain constant when the speed altered. Moreover, it is easily proved that, when output and speed diminish, the fuel consumption, so far from being lessened, is very considerably increased.

Grauert's diagram is reproduced in Fig. 32.

Now we must not suppose that in the production of this diagram three equal turbines of 1,500 horse-power were used, working on a single shaft parallel to the boiler. For to use a second and a third engine of 1,500 horse-power directly the required output exceeded 1,500 and 3,000 horse-power respectively would be too extravagant a method to be of any practical use.

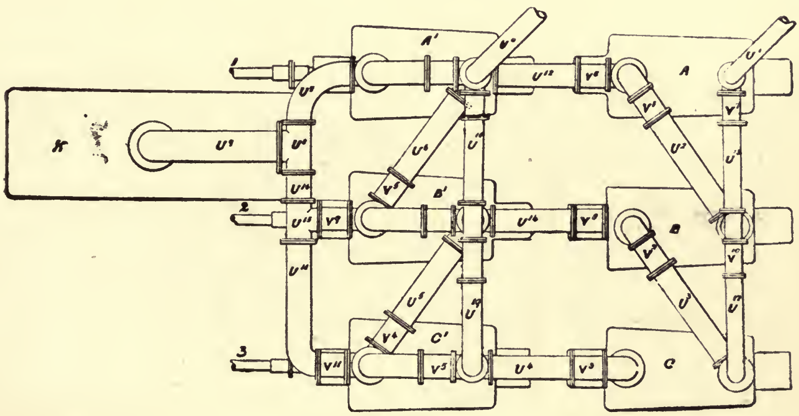


Fig. 31.

The experimental trials on which *Grauert's* diagram was based were, no doubt, made on a normal stationary Parsons turbine, in which it was possible to alter the rate of revolution without change of output. A very similar diagram is found on page 37 of a paper circulated by Parsons' representatives, "The steam turbines of *Brown*, *Boveri*, and *Parsons* for *stationary* and *marine engines*."

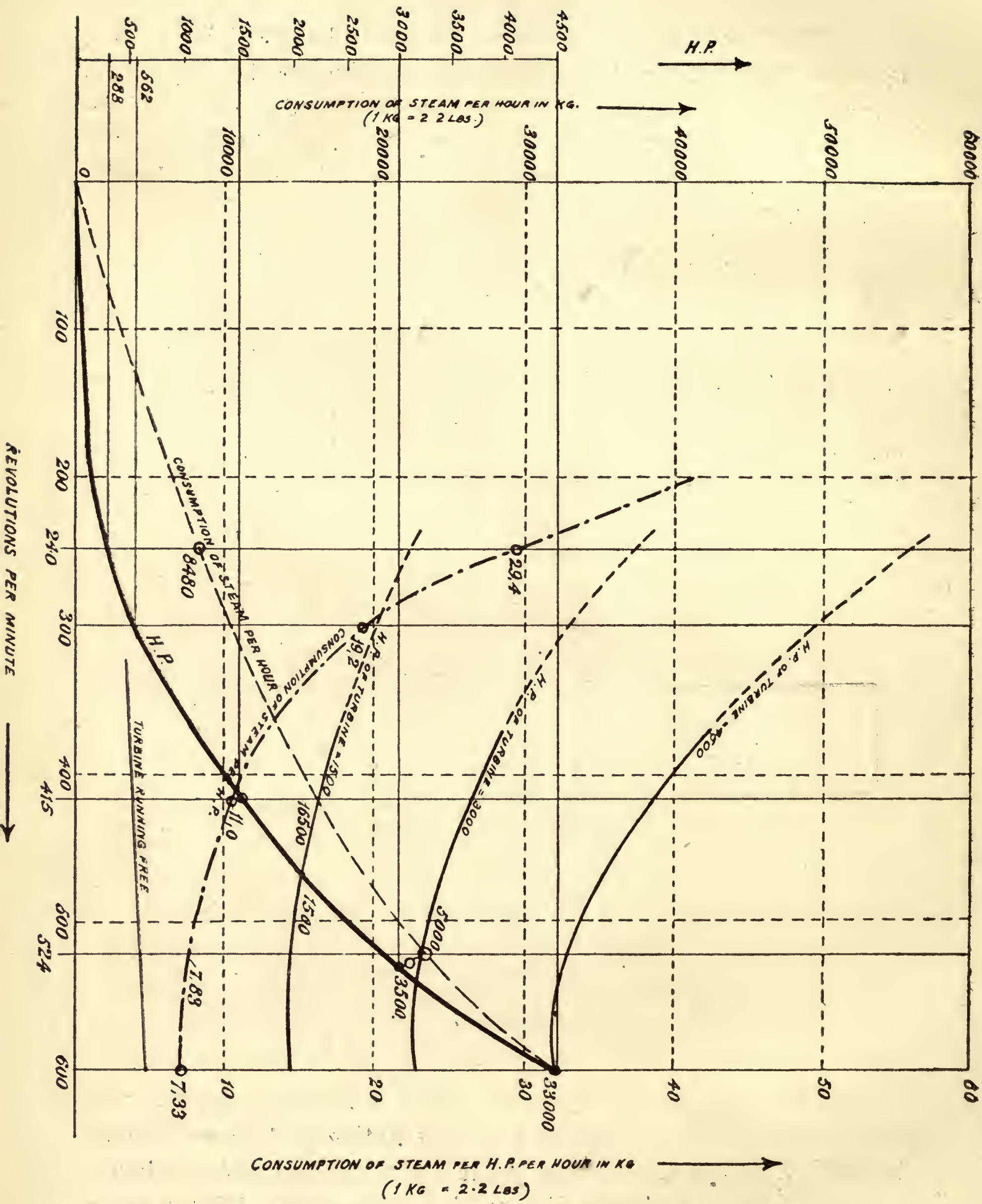

H.P.
CONSUMPTION OF STEAM PER HOUR IN KG.
(1 KG = 2.2 LBS.)
REVOLUTIONS PER MINUTE
CONSUMPTION OF STEAM PER H.P. PER HOUR IN KG
(1 KG = 2·2 LBS)
CONSUMPTION OF STEAM PER HOUR
CONSUMPTION OF STEAM PER H.P.
H.P. OF TURBINE = 1500
H.P. OF TURBINE = 3000
H.P. OF TURBINE = 4500
TURBINE RUNNING FREE
8480
29.4
19.2
11.0
7.83
7.33
16500
1500
3500.
5000
33000
288
562

Fig. 32.

The curve given here for light load is that used in *Grauert's* diagram. It is only applicable to marine engines with uncoupled shafts, and is worthless for trials of steam consumption on marine engines, so that on this ground also we assume that a stationary *Parsons* turbine is referred to. If this assumption be correct, the conclusions drawn from the diagram are of value only when applied to the driving of stationary engines, and not in the case of marine engines working at markedly varying speeds.

In ships' turbines it is impossible to maintain constant output with varying speed of revolution. In such engines the output varies as the cube of the speed of rotation; for example, if the speed be halved, the output will have only an eighth of its former value.

Now the diagram shows four outputs of a single turbine for speeds of revolution differing but little from each other. To arrive at a satisfactory conclusion we must consider the lower speeds also. The diagram, however, has a very different aspect when it is extended so as to cover the smallest ordinary speeds as well as the higher ones.

It is usual to take 40 % of the maximum for the lowest speed ordinarily employed. Since we may suppose the speed roughly proportional to the number of revolutions in a given time, the limits to be considered lie between 600 and 240 rotations per minute.

Now, in a marine engine the output (as mentioned before) varies approximately as the cube of the number of revolutions per minute, and the 600 revolutions per minute necessary for the maximum output of 4,500 horse-power with a steam consumption of 73,000 lbs. an hour (about 16·1 lbs. per horse-power per hour), being taken as a basis, the consumptions for other outputs work out as given in Table V., here following. We assume that in the diagram an ordinate of 1 mm. represents an hourly use of 1,100 lbs., or a consumption per horse-power per hour of 1·1 lbs.

TABLE V.

Output in horse-power.	No. of revolutions per minute.	Steam consumption.	
		Per hour.	Per horse-power per hour.
4,500	600	73,000	16·1
3,000	522	51,800	17·3
1,500	415	36,400	24·2
562	300	23,800	42·3
288	240	18,700	64·8

Grauert's diagram reproduced in Fig. 32 was extended with the help of this table. The curves thereby added show clearly the great difference of output and corresponding fuel consumption in one and the same turbine for greater and smaller speeds.

The trials on the well-known "*Turbinia*," constructed at the *Parsons* works, have given similar results. *Prof. Ewing* has collected these at the instance of the *Marine Steam Turbine Company*. They were published in a paper read by *Parsons* before the *Institution of Naval Architects* on June 26th, 1903, and are given, amongst other matters, on page 185 of the "*Marine Engineer*" *for 1903*. Table VI. contains these results.

TABLE VI.

Output in horse-power referred to the resistance of the ship.	Speed in knots.	Steam consumption per horse-power per hour		Steam pressure per square inch	
		in lbs.	in kg.	in lbs.	in atm.
980	32	23·7	13·0	162	11·0
704	28	33·0	15·0	129	8·75
325	20	41·9	19·0	72	4·9
184	16	52·5	23·8	47	3·2
90	13	68·3	31·0	29	2·0
34	10	88·2	40·0	13	0·9

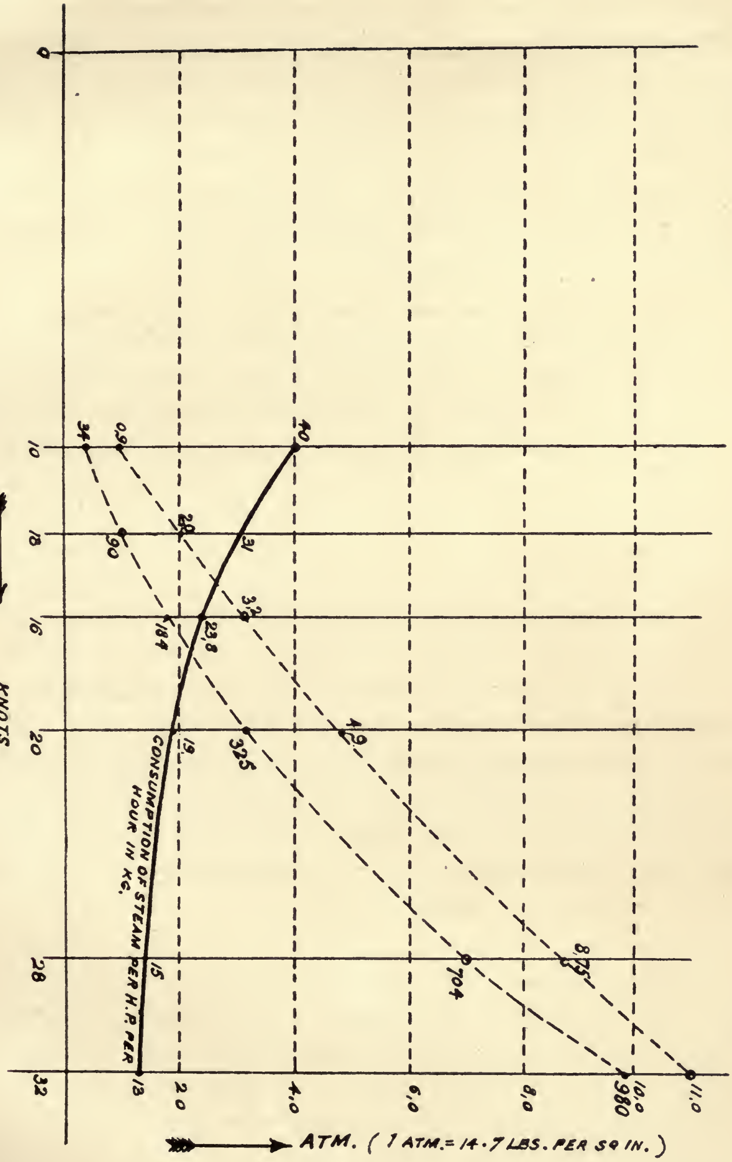

KNOTS
ATM. (1 ATM.= 14·7 LBS. PER SQ IN.)
CONSUMPTION OF STEAM PER H.P. PER HOUR IN KG.
0
10
16
20
28
32
2.0
4.0
6.0
8.0
10.0
11.0

Fig. 33.

In the last column of this table is given the initial pressure of the steam at its entrance into the turbine. We see that when the output is at its minimum the steam must be throttled below atmospheric pressure before it finds its way into the engine. This explains the great losses of steam sustained by these turbines as their output diminishes. Fig. 33 gives the results of Table VI. in graphical form. Comparison of Figs. 32 and 33 shows the similarity of the two *Parsons turbines*, and proves that when they are used for ships the steam consumption increases considerably as the work done per horse-power per hour diminishes.

It is now quite evident without further explanation, why the torpedo-boat destroyers "Viper" and "Cobra" consume nearly twice as much fuel on an ordinary voyage as their sister-ships "Albatross," &c., which are fitted with reciprocating engines. The turbines of the "Viper" and "Cobra" are shown in Fig. 30 (Fig. 2 of the patent No. 103,559). Since it was mainly on this patent that Parsons relied in his lawsuit against *Schulz*, it is sufficiently shown that it is not by this method that we can solve the problem of securing the same economy with small outputs as with large ones.

Parsons also has recognised the fact that the arrangement of his patent No. 103,559 cannot bring much success. This is evident from the circumstance that he had recourse to reciprocating engines for slow speeds. This system was protected in the English patent 16,551/1900, but also proved a failure. The German Naval Administration, for example, was averse to a composite system, and *Parsons* then had recourse to detachable turbines on the *Schulz* pattern.

Schulz had patented his system in England on April 23rd, 1901. Soon after the publication of the patent specification (8,378/1901), *Parsons*, on August 7th, 1902, brought out the English patent 17,391/1902, which relied on the principle of small detachable turbines for ordinary voyages, similar to those in *Schulz*' previous patent, 8387/1901. *Parsons*' only alteration was the introduction of additional steam into the individual turbines, by which he

obtained a better graduation of the output while sacrificing the advantage of greater economy.

It is true that *Parsons* described in his patent, and fitted in the German torpedo-boat "S 125," and in the English cruiser "Amethyst," a system in which the smallest turbine is cut off during fast voyages

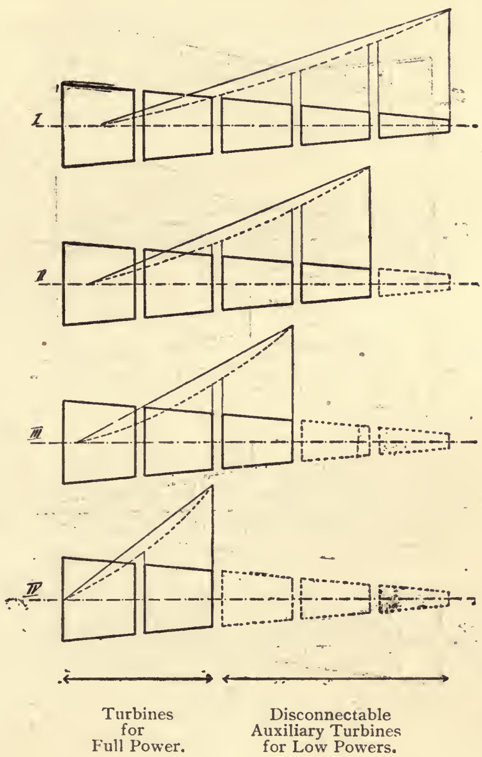


Fig 34.

and the steam led directly into the second turbine (Fig. 37). In this he followed the *Schulz* system. He failed, however, to notice that, owing to the distribution of auxiliary turbines adopted, the outer shafts communicated very unequal amounts of energy to their screws, and that, consequently, the system is quite unsuited for marine purposes.

These considerations quite dispose of *Parsons'* statement, that he had by the method of his former patent No. 103,559, practically solved the difficulties arising from the varying rates of speed at which marine turbines must work. This will be still clearer if we submit the specification of the patent to a searching examination.

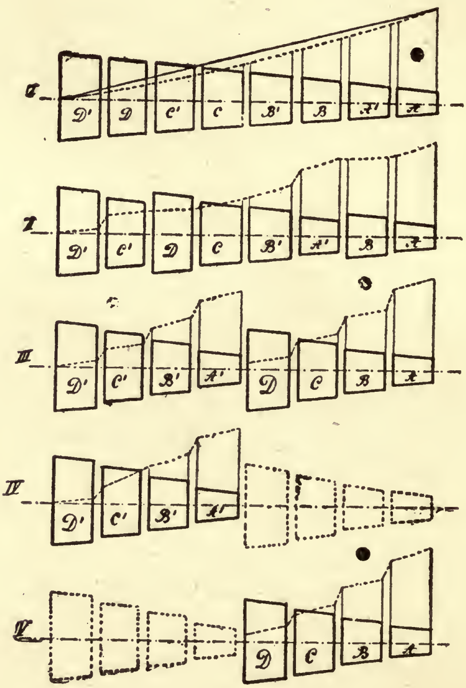


Fig. 35.

We can only obtain the maximum economy with a turbine, or with a combination of turbines, when the expansion of the steam from the boiler to the condenser has a considerable range, and when the current in the turbine flows without discontinuous changes of pressure. For this purpose we must give a suitable succession of cross-sections to the passages by which the steam is led from its entrance into the first turbine to its exit into the condenser. To

emphasize the difference between *Schulz'* and *Parsons'* turbines, we reproduce a number of diagrams in which, for the sake of clearness, all the turbines of the plant are placed side by side. The diagrams are only intended to show the variation of pressure as the steam passes through the turbines, and make no claim to accuracy in detail.

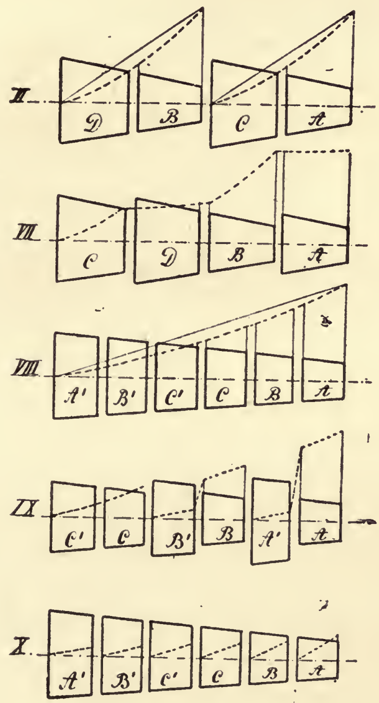


Fig. 36.

From Fig. 34 it is quite clear that *Schulz'* system satisfies the abovementioned conditions. As we pass from the maximum output to a lesser one and so down to the minimum, we are continually adding smaller and smaller turbines. In all cases the steam passes at its boiler pressure into the foremost of the additional turbines, and undergoes a continuous and complete expansion as it passes through passages of continuously increasing cross-section.

Now we cannot construct a turbine plant in such a manner that at every speed all the working-wheels contribute to the expansion of the steam. It is usual to secure the most favourable case when the output is at its maximum, as is apparent from the diagrams. As the output is lessened, an increasing number of turbines at the condenser end cease to share in the work. This is, however, inevitable, and does not affect the general principle, *that increase in the number of turbines should accompany decrease of output.*

Parsons has attempted to secure an economical use of fuel at all speeds by *a redistribution* of the turbines, but has only actually attained this in those cases which in Figs. 35 and 36 are numbered I, VI, and VIII. All the remaining arrangements fail to secure a continuous expansion of the steam. It is apparent from the patent specification that *Parsons* also, for small outputs, sends the steam through all the turbines one after another, but it appears that for *large* outputs he also sends it through *all the turbines* by a route that is several times as long. If we consider first of all the method of Fig. 35, we see that, on plotting the fall of pressure, we get a normal curve only when all the eight turbines are placed in series. In II. the turbines B and D hardly contribute any share to the output, while between A′ and B′ and also between C′ and D′ we have an exceptionally large fall of pressure without any corresponding performance of work. Fig. 36 shows the decrease of pressure due to the arrangement of Fig. 30 (Fig. 2 of *Parsons*' patent). This is the system fitted on the "Viper" and "Cobra." If we assume that the turbines work satisfactorily when connected in parallel (Fig. 36, VI.), this can no longer be the case when all four of them are connected in series to suit small outputs. The turbines A and D then do hardly any work—a fact which explains the large coal consumption of these vessels. Consideration must, moreover, be given to the above-mentioned inequality in the distribution of the work.

Similarly, Figs. 36, VIII, IX, and X, prove that the system of Fig. 31 (Fig. 3 of the patent specification) is economical only when the turbines are connected in series.

Except, then, in the case of the distributions I, VI, and VIII we cannot obtain a gradual fall of pressure by any of the methods of the patent No. 103,559. The steam passages leading from the boiler to the condenser will not have properly graduated cross-sections, and the division of work among the screw shafts will be so uneven, that the distribution methods described in the patent specification cannot possibly produce a plant that will be economical at all speeds.

For the principal turbines of a plant on the *Schulz* system (Fig 34) intended to drive a large-sized man-of-war, from 60 to 90 working-wheel blade-rings are necessary. The number of rings in the auxiliary turbines depends on the smallest output required. In most cases a distinctly larger number of rings is necessary for the auxiliary than for the main turbines. If we have found the number of blade-rings required in the principal turbines, and also the number in the other turbines suitable for the smallest reasonable speed of working, we can evidently get every possible speed that lies between the maximum and minimum limits. We have only to determine how best to group the blade-rings of the auxiliary turbines so that the various speeds may be obtained with the greatest economy.

In Fig. 34 is shown an arrangement of this kind with three detachable turbines. For the smallest output all three groups of blades are employed. The steam passes through each in turn, then through the main turbine, and so on, into the condenser. It may happen that the steam has expanded completely before it has passed the last blade-ring. This last ring will then turn in the vacuum without performing work, but the slight disadvantage connected therewith is inevitable

If a greater output be required, the first auxiliary turbine is cut off, and the steam enters the second one. For still higher outputs only one auxiliary turbine is used, and for the maximum only the main turbine.

By this means we can get the best result at every intermediate speed, provided we know how to determine the total number of rings necessary for the smallest output required, and how to divide them properly into the various groups.

If, now, an output having the highest possible economy be desired for only one or two given speeds besides the maximum, it is natural to apply the *Schulz system* only so as to satisfy these require-

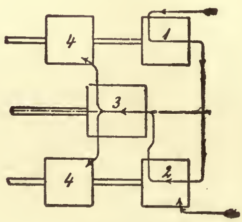


Fig. 37.

ments and to pay no attention to the intermediate outputs. This, however, in no way detracts from the merits of the new method of working, the inventor of which is *Schulz*.

It is, moreover, a matter of complete indifference, whether his auxiliary turbines are mounted on the same shaft as the main turbine, or whether they are distributed over several different shafts and connected by pipes of greater length. *Schulz* has given due consideration to such arrangements, and has applied for further patents.

After the publication of *Schulz'* new system, *Parsons*, as mentioned above, employed similar arrangements. For instance, in a French torpedo-boat he placed a turbine for ordinary voyages in front of the main turbine, and in the English cruiser "Amethyst" and the German torpedo-boat "S 125" he built a similar auxiliary turbine in front of the main turbine of each of the outer shafts.

Fig. 37 shows this arrangement. The two auxiliary turbines are denoted by 1 and 2: the three main turbines by 3,4,4. When the output is at a minimum, the steam goes from 1, through 2, to 3. Then it is divided and passes through 4 and 4. If a larger output be required, *Parsons*, following *Schulz'* method, cuts out 1, and sends the steam directly into 2, and thence, by way of 3, to 4 and 4.

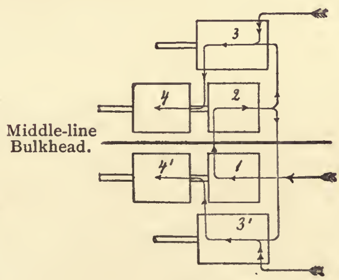


Fig. 38.

For turbine plants with four screw shafts, *Parsons* solves the problem in a similar manner. Fig.38 shows the system adopted on the small German cruiser "Lubeck." The usual voyage speeds of this vessel are 11 and 19½ knots per hour, corresponding with outputs of 1,400 and 7,000 horse-power respectively.

When the smaller speed is required, the steam passes first into the auxiliary turbines 1 and 2, placed on the two inner shafts in front of the corresponding main turbines. On leaving 2 it is divided and passes through the main turbines 3 and 4 on one side of the ship and 3′ and 4′ on the other side. For high speeds and for some of

the intermediate ones 1 and 2 are cut off, and the steam passes directly into 3 and 3′.

In all these cases, *Schulz'* new principle of detachable turbines is adopted, but only to a limited and, therefore, imperfect extent.

Moreover, both the plants of Figs. 37 and 38 give rise to uneven and asymmetrical distributions of the work amongst the shafts. Besides, in the system of Fig. 38 the central partition-wall of the ship is pierced three times by steam pipes, including one pipe for the go-astern engine. This interdependence of their two sides is certainly unfavourable for the working of the engines.

We have shown that in *Schulz' Marine Turbine* (Patent No. 168,863), there is a continuous fall of pressure and, therefore, a perfect use of the available energy *under all possible circumstances.* In *Parsons'* system, on the other hand, this is only true for *certain special cases. Schulz'* method certainly gives the surest solution of the problems connected with the turbine engines of war and merchant vessels.

When the output of a reciprocating engine alters, the economy does the same, for it diminishes with diminishing output. The same is true in steam turbine engines also.

Machines for stationary plant almost invariably work at speeds bearing constant ratios to the different outputs. These ratios vary within narrow limits, being mostly either 1 : 2 or 1 : 4. On this speed of working the economy of stationary engines chiefly depends. The case is different with marine engines, and especially with those of men-of-war. Here the speed may vary from the maximum down to the 1/15th part of it.

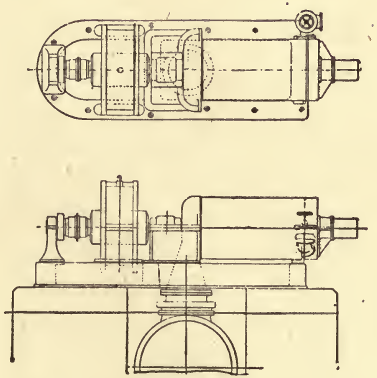
In steam turbines the cross-section of the steam passages is made just large enough for the highest output, *i.e.*, for the maximum consumption of steam. When the work to be done is smaller, these passages are much too large. Since a resort to throttling is here necessary, the steam pressure in the first expansion stage must fall abruptly, and waste of energy is always the result. Now *Schulz* has, above all things, secured the maximum of efficiency at different speeds by dividing the whole turbine complex into detachable portions and by passing the steam into these various turbines in such a manner that a state of expansion is always present. When the output decreases, an increased number of turbines or turbine drums come into play. For the minimum of work all the turbines share in the propulsion.

The paradoxical nature of the arrangement makes it the less surprising that this device should have escaped the notice of the earlier designers of marine turbines, *Parsons* and *Rateau*. *Schulz* was the first to perceive that here was the means of obtaining a steady fall of pressure.

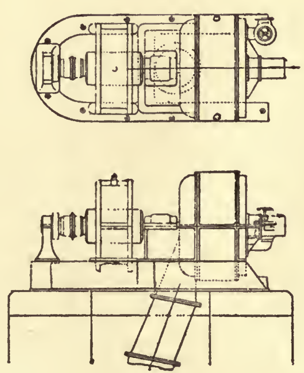
Now it is in general true, that every turbine makes the maximum use of the energy supplied to it only at one particular peripheral velocity and rate of steam flow. *Schulz* has accordingly chosen the number of expansion stages, the blading, and the cross-section of the steam passages in the individual turbines to suit a certain mean speed of flow. Since the elasticity of the steam has also to be considered, the loss of economy is very slight, even if the rate of flow vary within small limits. The number of auxiliary turbines necessary for maximum economy *during manœuvres* can only be determined by experience.

The *Schulz turbine* works at every speed with nearly constant boiler pressure. If its speed is to be reduced, the number of the auxiliary expansion stages is increased and the rate of flow of the steam becomes smaller, because the latter has to go through turbines

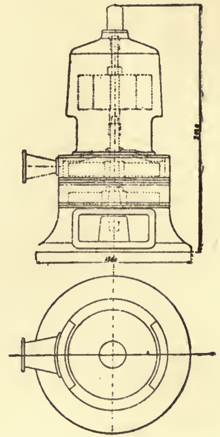
Schulz Patent Excess-Pressure Turbine.
H.P. 500-600.
Revol. per min.=3000.



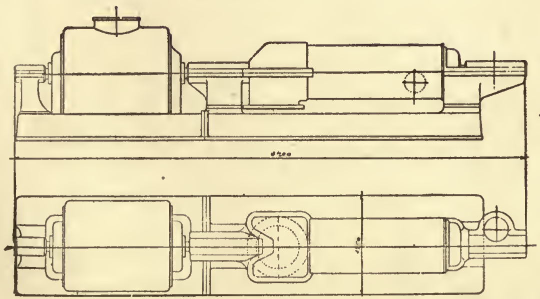
Schulz Patent Pressure Turbine.
H.P. 500-800.
Revol. per min.=2000.



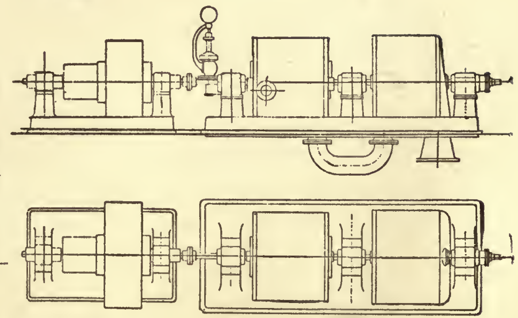
Curtis Turbine.
H.P 500-800.
Revol. per min.=2400.



Parsons Turbine.
H.P. 500-600.
Revol. per min.=3000.



Zoelly Turbine.
H.P. 500-600.
Revol. per min.=3000.



Figs. 39, 40, 41, 42, 43.
Scale, 1 to 100.

with passages of small cross-section before it passes into the main turbine on its way to the condenser. The speed of the turbine decreases as the number of the expansion stages is increased, and extravagance in fuel is thus avoided.

Now, it is impossible to secure an absolutely perfect use of steam at every speed.

If, for example, the main turbine be so designed that the steam will, at the highest output, already have reached the condenser pressure at its exit from the last expansion stage, it is inevitable that, when all the auxiliary turbines are used, the last blade rings of the main turbine should rotate without performing work.

In fact, the condenser pressure is attained before the steam leaves the main turbine; for it is impossible so to arrange the blading and the cross-sections of the passages that all requirements are satisfied at both high and low pressures. If, at the smallest output of the plant, the last few expansion grades be ineffective, the uselessly revolving hindmost blade rings waste work in unnecessary ventilation, and the amount of this must be deducted from the effective output of the engine. The loss is, however, small, owing to the fact that the condenser pressure is always low, and it also becomes less as the rate of steam consumption diminishes.

We have shown that the *Schulz turbine system* possesses the following essential advantages for marine use :—

1. The maximum of economy at all speeds.

2. High pressure of the steam on its entrance into the turbine at every speed.

3. Powerful and prompt action in reversing.

4. The regulation of the end thrust exerted on the shaft by the steam and by the propeller respectively.

5. The combination of action and reaction turbines in one system.

6. The division of the whole plant into several detachable portions, so that a steady expansion of the steam is always attained.

7. The simple and convenient arrangement of the various valves.

8. An even distribution of the work over the different shafts.

9. Economy in space and weight in comparison with other systems (Figs. 39 to 43.).

In Figs. 39 to 43 a comparison is made between the space requirements of the best known turbine systems. The diagrams show the plans and elevations of these turbines on a scale of 1 to 100. The outputs range approximately from 500 to 600 horse-power—in the cases of *Curtis'* turbine and *Schulz'* reaction turbine from 500 to 800 horse-power. The speeds of revolution are from 2,000 to 3,000 per minute. This comparison shows that the *Schulz turbine* economizes space to a considerably greater degree than do those of the other systems.

The patent specifications bear reference to the further development of the *action turbines*, as well as of the *reaction turbines* on the *Schulz system* and to the *distribution* of the *turbines*, in a *compound engine*, *over several shafts*.

In this, special attention is paid to the simplicity and convenience of the levers which control the cutting off of the various auxiliary turbines during manœuvres.

Unlike many other inventions on the domain of steam turbines, the *Schulz system* has a firm foundation in the extensive practical experience of its designer. It is to be greatly desired that these turbines be soon brought into competition with those of other types; their success will then not be long delayed.